Power and The Engineer

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IN GIVING his reasons for changing his vocation, an engineer with long years of experience says, "I have been connected with this business for half a life time and am retiring discouraged and disgusted.

"Conditions are getting worse all the time and the practical man who depends on his knowledge of the business and his merits to hold his job without a 'pull' is going down and out. He must make way for the paper and pencil engineer with big ideas, who shows on paper how to run boilers and engines without burning coal. It is too much for me and I am done with it forever."

In a heated political campaign some years ago a strong, earnest man, expressing his contempt of the half-hearted efforts and his disappointment in the tactics of a weaker brother intrusted with an important mission, said, "God Almighty hates a quitter."

There is honor in honorable defeat, but a failure to conquer the unconquerable should be only a spur to the man who takes it rightly. It should stir his spirit and urge him on to redoubled effort in the next battle.

It is a safe bet that if one has never made a failure he has never scored a success worthy the name.

What does a real knockdown mean? It means a chance to rest while taking the count, a chance to spit on the hands and come up with the determination to do better next time.

The chief factor in success is patient, dogged industry that neither flinches nor falters—the unconquerable courage that will never submit or yield. It is success itself.

There are many successful engineers in the world, with perhaps no two alike in physical endowment and mental attainment, but all possess this one moral quality—a grim tenacity of purpose to hang on and hold out through everything until the desired end is reached.

"If you have never met more than your match, it is likely you have seldom toed the scratch."

Modern Manufacturing Power Plant

By Warren O. Rogers

In this turbine plant the arrangement of the machinery is ideal. Nothing is cramped, and the plant is light and well ventilated. The method of handling the coal and ashes is convenient and up-todate. A turbine-driven centrifugal pump is used as a boiler feeder.

If an engineer were to step into a power plant and find in one large, light, well ventilated room steam turbines and generators, air compressors, hydraulic pumps, apparatus for a hot-water heating system, exciter units and refrigerating machines, all delivering their output to neighboring manufacturing buildings, he would doubtless exclaim, "a combination central manufacturing power plant." The following deals with such a plant:

The location is ideal, being close to the main line of a certain railroad; it is also but a short distance from a navigable river. The building is erected on a plot of land practically free from other buildings; the engine room is well lighted and the ventilation is all that can be desired, due to the large adjustable windows in the side walls of the engine room. A concrete roof supports two monitors which have adjustable windows extending their entire length. All of these windows are adjusted from the floor by means of handwheels and extension rods. The room is so spacious that the heat from the various machines is hardly noticeable and within it is comfortable even on a hot summer's day. Fig. 1 presents an excellent view of the generator room, and shows the general arrangement of the machinery.

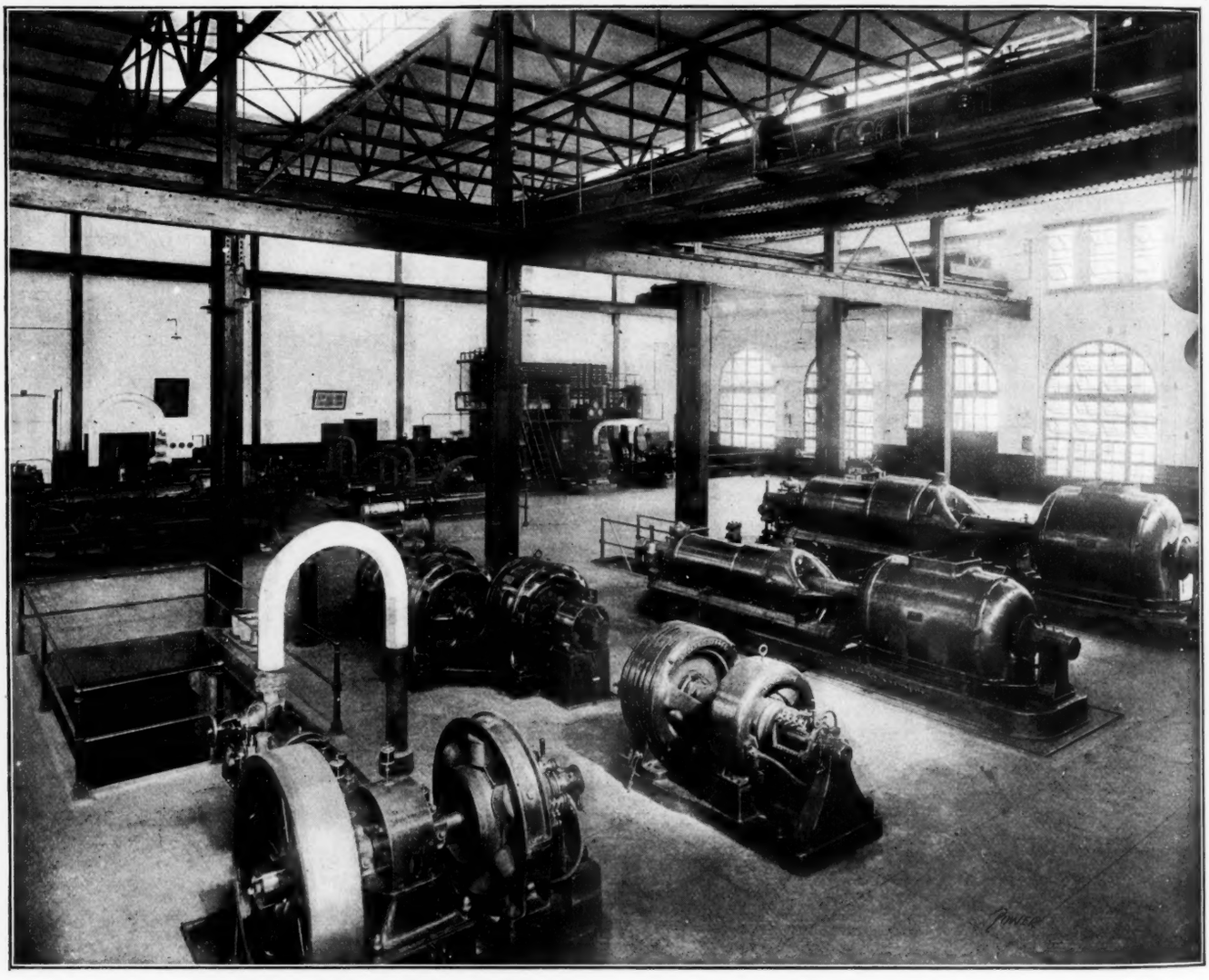


FIG. 1. MAIN PORTION OF TURBINE ROOM, SHOWING GENERAL LAYOUT OF PLANT

COAL DELIVERY

Fuel is delivered to the yard in coal cars and is dumped either into the conveyer tipple or in a pile under a trestle to serve as a reserve supply. Fig. 2 shows the coal and ash tipple. A car of coal is on the scale ready to be discharged into the coal tipple, which is constructed similarly to the ash tipple located above.

When coal is desired in the coal bunkers, which are located above the boilers, the link-belt conveyer, which is 596 feet long and contains 298 buckets, is started by throwing in a switch located in the boiler room. The crusher rolls, which are set below the coal chute shown in front of the concrete coal tipple, are also started at the same time. As the coal chute leading to the coal-crushing rolls is constructed with a movable bottom, having a forward and back or shaking motion, the coal is automatically fed to the crushing rolls.

After the coal is crushed it is elevated to the top of the bridge and carried to the concrete coal-storage bin, which extends the entire length of the boiler room. This

storage bin is divided into six compartments, each capable of holding 250 tons of coal. Two delivery outlets lead from each of these compartments, the delivery openings being in the center of each boiler and directly over the dutch-oven furnaces. When a stoker requires more fuel a Standard coal scale, which runs on a suspended track, is pushed under the coal-supply outlet and the coal is weighed a second time, and then discharged to the stokers. Each row of boilers has a separate coal-supply chute and scale.

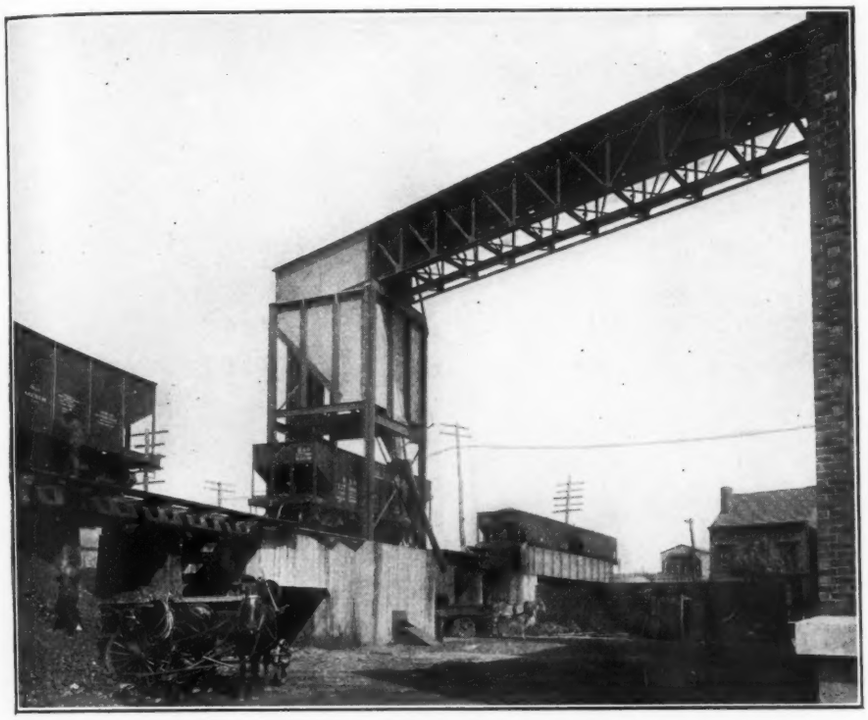


FIG. 2. COAL AND ASH TIPPLE, SCALES AND CONVEYER

BOILER ROOM

There are six 500-horsepower Babcock & Wilcox boilers, carrying a steam pressure of 160 pounds per square inch. They are set in three batteries of two boilers each, and each boiler is equipped with two Murphy stokers. The stokers of each boiler are driven by a small steam engine, as shown in Fig. 3, which also presents a view of one side of the firing alley. The boilers are of the three-drum type and the brick setting of each battery is covered by a metal casing. This prevents air leakage through the setting, and a higher efficiency is obtained from the furnaces than would be possible with the ordinary setting.

The massive supporting beams of the overhead coal bin are shown in Fig. 3, also the arrangement of the boilers and weighing machine. The flooring is made of concrete and as the ashes are handled below the boiler-room floor the place is easily kept clean.

At the rear of each battery of boilers is placed a Green economizer. Fig. 4 shows the arrangement of the two economizers that are set behind the boiler shown in Fig. 3. The tube scrapers on both economizers are operated by one motor by means of a belt drive. A bypass flue is built beneath the superheaters so that either one or both can be cut out of the circuit of flue gases passing to the stack. This is accomplished by means of dampers, as shown. The gases from one or both batteries of boilers can be passed through one or both of the economizers, and so far the two economizers have been utilized.

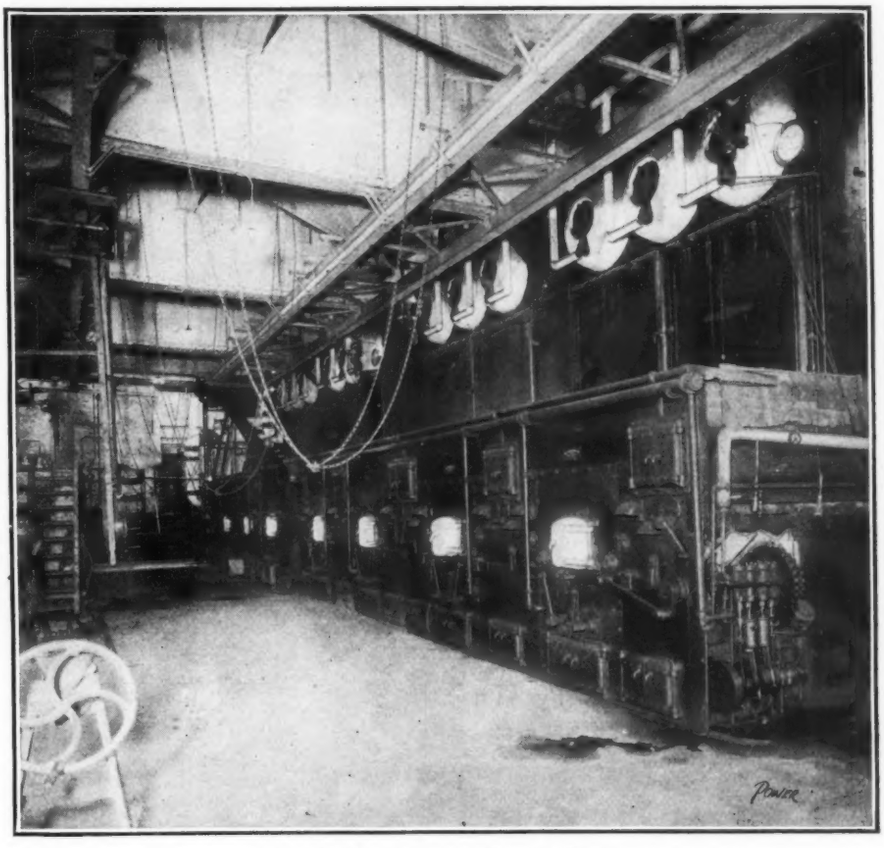


FIG. 3. VIEW OF THE BOILER ROOM

All feed water is heated in a Hoppes open heater to 200 degrees and leaves the Green economizer at a temperature of from 245 to 265 degrees Fahrenheit. It is supplied to the boilers by two pumps: one a Cameron simplex steam pump, which is held in reserve; the other a D'Olier centrifugal pump direct coupled to a Terry turbine. These units are shown in Fig. 5, with the simplex pump in the foreground. The centrifugal pump in a 24-hour run averages approximately 10,000 cubic feet of water per hour, the night consumption being about one-third of the amount required during the day.

A main feed pipe to each battery of boilers is run above the steam drums, and branch pipes drop down and connect to the three drums of each boiler. The feed water is regulated by hand by means of an extension rod from the valve wheel extending down to within reach of the firemen.

DISPOSAL OF ASHES

Ash disposal in some plants is an item of considerable expense. Any arrangement that is efficient, and at the same time simple to operate, should be a success. These two features are found in the ash-handling system in this plant, as will be

apparent from the view of the ash-conveyer tunnel, shown in Fig. 6. This tunnel extends the entire length of the boiler house, centrally between the two rows of boilers, to the coal tipple.

and the chute lifted and secured in position, as shown at the further end of Fig. 6. As the sides of the elongated hopper are lined with sheet iron, no difficulty is encountered from the ashes "hanging

are grouped at one end of the generating room. The larger turbine is of 1000 kilowatts capacity and is used to furnish the electrical current for manufacturing purposes. The smaller turbine is of 500

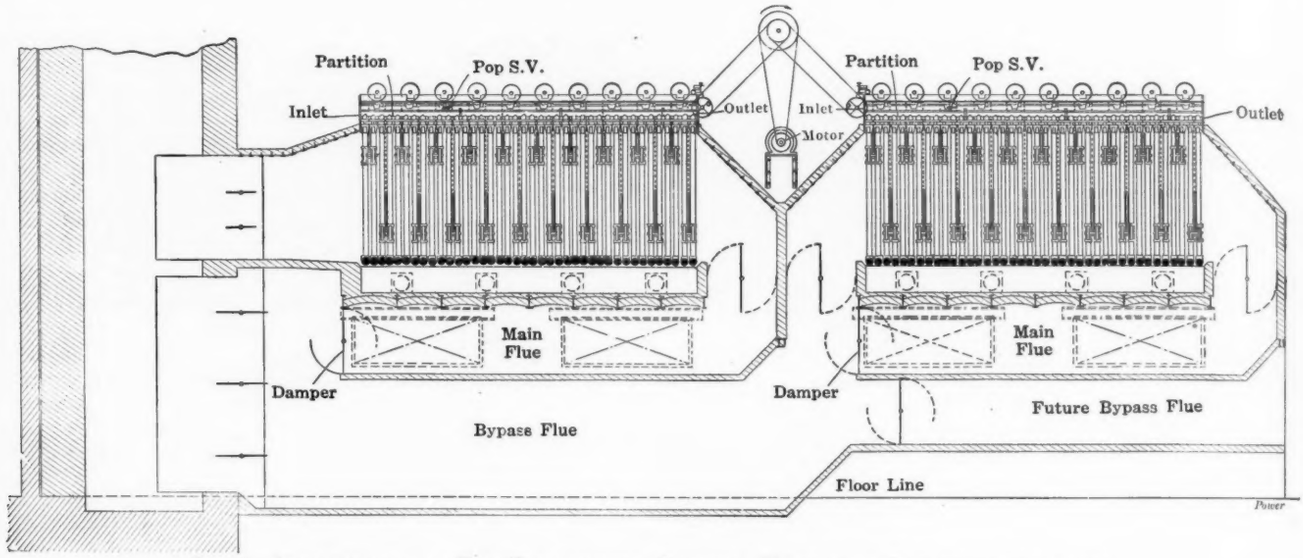


FIG. 4. ARRANGEMENT OF TWO ECONOMIZERS, SHOWING METHOD OF BYPASSING THE GASES

The ashpit under the grate of each boiler is constructed with a bottom sloping at an angle of 45 degrees. The outlet of each ashpit leads directly into the conveyer tunnel. In the foreground two of the ashpit doors are open and the

up." The same conveyer buckets are used for handling both coal and ashes.

TURBINE ROOM

Stepping from the boiler room into the generating room, one is almost amazed

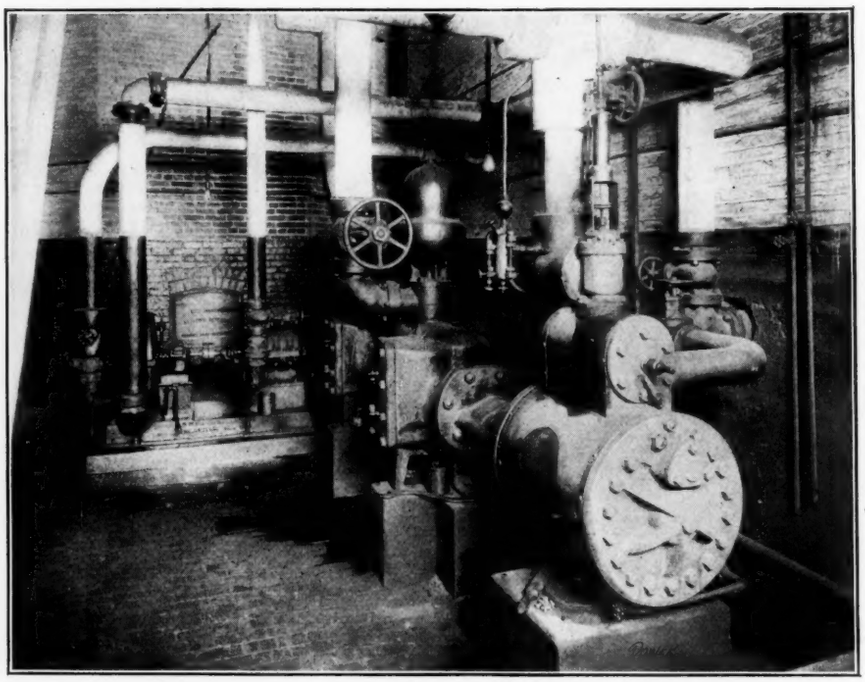


FIG. 5. BOILER-FEED PUMPS

chute dropped ready for the ash man to hoe the ashes out into the moving hopper buckets which convey it to the upper concrete tipple, shown in Fig. 2. Here the ashes are dumped and are loaded from time to time into a wagon placed under the delivery chute. When an ashpit has been cleaned the door is closed

at the space devoted to this part of the plant. Nothing is cramped, and every machine is set with order and precision. The various types of the same machine are grouped and arranged in a manner not easily improved upon.

Referring to Fig. 1 again, it may be seen that the electrical generating units

kilowatts capacity and is used to carry the night load. Next to the iron supporting columns is the De Laval turbine of 200 kilowatts capacity. This unit is held as a reserve, but can be coupled to the 500-kilowatt unit in case the large turbine generator should become disabled. The two Allis-Chalmers turbines are run condensing, but the De Laval turbine is run noncondensing.

In the foreground a motor-driven direct-coupled exciter unit is shown, also a steam-driven direct-coupled unit used for the same purpose; the latter, however, is held in reserve. At the generator end of the turbines, not shown, is the switchboard from which the electrical output of the plant is controlled.

Over these machines is a 25-ton Niles motor-operated traveling crane for lifting the heavy parts of these units when necessary.

A near view of the refrigerating machines is shown in Fig. 8, also the air compressors and hydraulic pumps. There are two York refrigerating machines, each of 30-ton capacity and of standard make. One feature worth noting is the arrangement of the condenser coils. There are two sets of double pipe coils, but either one can be used with either machine, or both used with either of the machines. Steam is delivered to the steam cylinder from the vertical pipe which connects with a main steam pipe in the basement.

A motor- and one steam-driven air compressor furnish compressed air at a pressure of 70 pounds per square inch. The air is conveyed to the factory buildings through a pipe placed in a tunnel, which also accommodates the main feeder wires, hot- and cold-water pipes, live-steam and all return pipes. Weather con-

ditions have no effect upon the distributing end of the plant. The belt-driven air compressor receives its power from a 50-horsepower Bullock induction motor located in the basement. The motor-driven unit is used during the summer months, and the steam-driven unit during the winter months when the exhaust steam can be utilized in heating the water used for heating the shops.

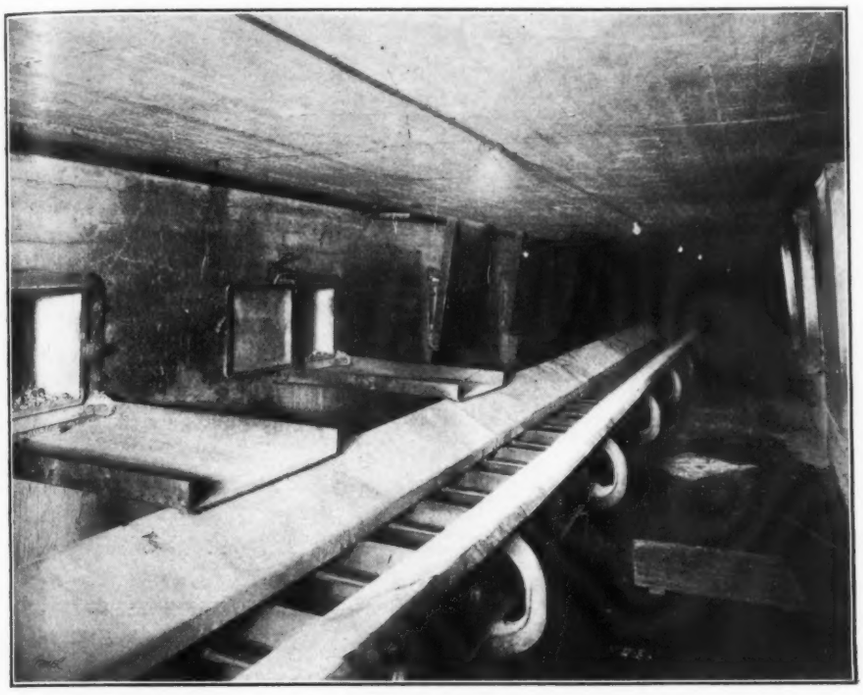


FIG. 6. METHOD OF DISPOSING OF ASHES

HOMEMADE ACCUMULATOR

Between the air compressor and the side wall are two hydraulic pumps which work against a pressure of 2500 pounds per square inch. A homemade accumulator weight keeps the pressure in the delivery-pipe lines constant. The construction of this weight is shown in detail in Fig. 9. It weighs 95,000 pounds and is made of cement in which scrap iron and broken sections of concrete sidewalls are mixed. It is made in sections, two to each layer, with the exception of the bottom layer which is made solid. When constructing the weight a sheet of tar paper was placed between each layer and section, which allows the removal of any section without breaking the neighboring one. Eye bolts are set in each concrete slab below the top surface, which makes it an easy matter to lift them with the crane. These slabs are each 4 feet 3 inches wide, 8 feet 6 inches long and 2 feet thick. These, together with the solid, bottom slab, make a weight 8 feet 6 inches square and 8 feet 10 inches high, consisting of nine blocks of concrete. The concrete of each block was made of a one, four and seven mixture of cement, sand, scrap iron and old concrete. This weight cost in round numbers $250. Had it been made of cast iron at 1½ cents a pound (as quoted), the cost would have been $1425; therefore a saving of $1175 was made, and, although occupying a little more space, the concrete weight answers the purpose exactly as well as a more expensive cast-iron weight.

The weight surrounds a central steel chamber in which the water is confined, and as the pressure increases, the weight is raised. The inlet connection from the pump is at the bottom of the column. A 6x6-inch wooden guide frame which has a cross piece fitted to the top steadies the weight. On each end of this cross piece three rollers are fitted. A wide-face pulley rolls on the edge of a strip of 3x⅜-inch tee iron; the other two rollers bear against the sides of the tee iron.

Attached to a top and bottom corner of the concrete weight are two angle irons. A cable is connected to the top angle iron by means of a turnbuckle. This cable extends down to a flanged pulley, and, after being wound twice around it, is extended to another flanged pulley attached to the throttle-valve stem of the steam pipe feeding the steam hydraulic pumps. The cable passes back to the first pulley and, passing over the top, is secured to the bottom angle iron. As the concrete weight rises and falls the valve is opened and closed.

CONDENSERS

The condensers for the two condensing turbines, Fig. 7, are of the jet type, and were made by the Allis-Chalmers Company. The 1000-kilowatt unit condenser is equipped with a Connersville Blower Company's rotary cycloidal vacuum pump. It

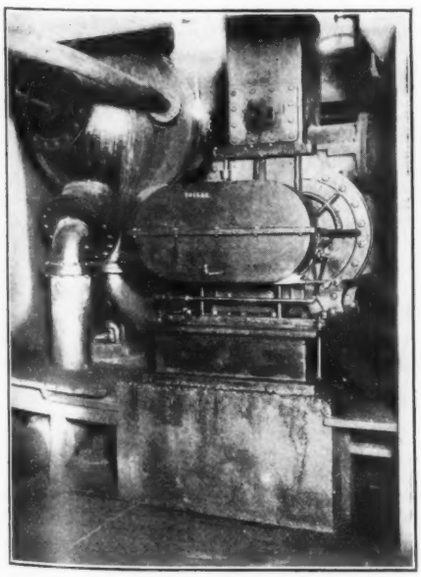


FIG. 7. ROTARY VACUUM PUMP AND JET CONDENSER

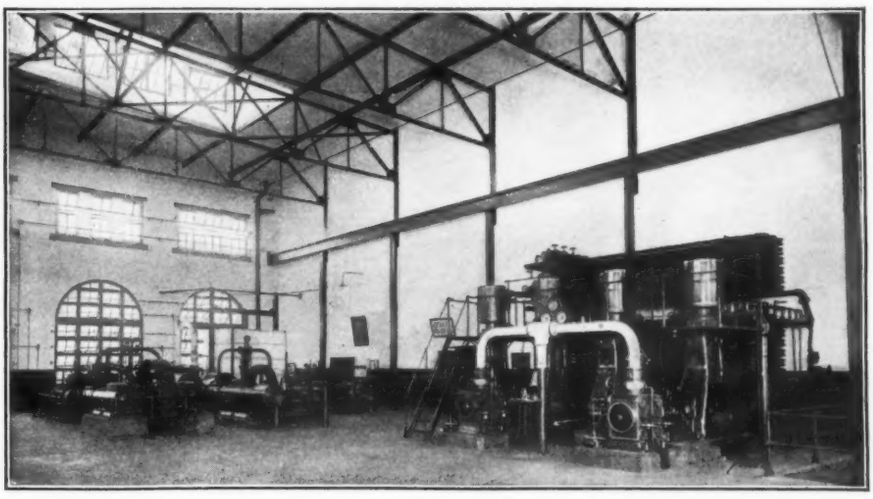


FIG. 8. AIR COMPRESSORS, HYDRAULIC PUMPS AND REFRIGERATING MACHINES

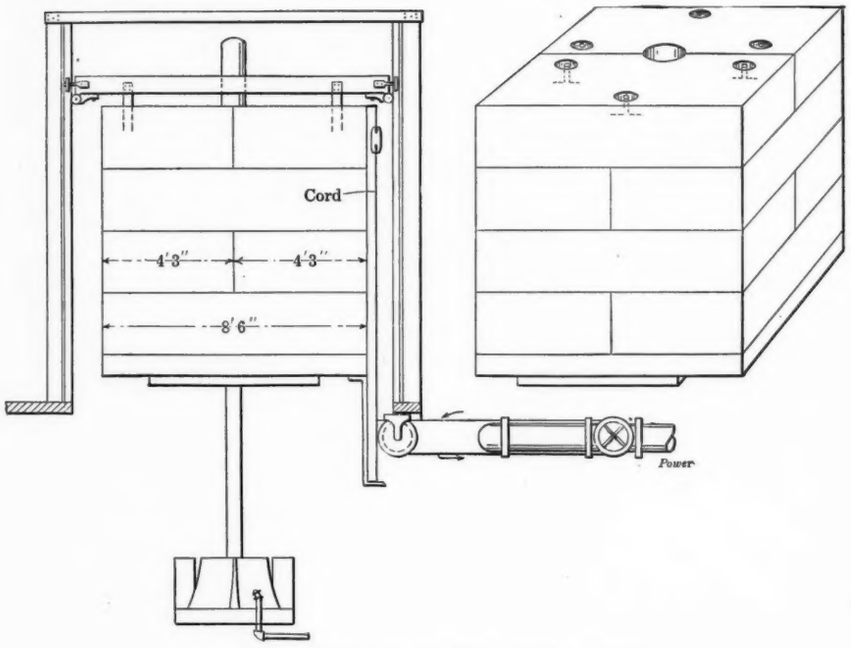


FIG. 9. DETAILS OF ACCUMULATOR WEIGHT

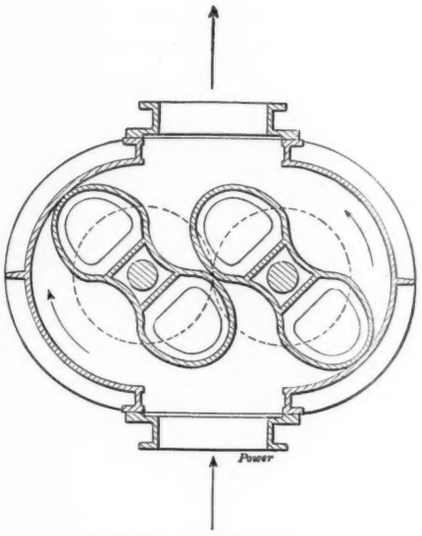


FIG. 11. CONSTRUCTION OF VACUUM PUMP

is driven by a piston-valve engine, direct connected, and runs at a speed of 80 revolutions per minute. The displacement is 35½ gallons per revolution. The arrangement of the vacuum pump on its foundation is shown in Fig. 10.

The 500-kilowatt turbine is fitted with a jet condenser; the same make of air pump is used as with the large machine. The pump is coupled to a piston-valve engine which runs at 95 revolutions per minute. The displacement of the pump is 13.10 gallons per revolution.

A sketch of the interior construction of the vacuum pump is shown in Fig. 11. The outline of the two so-called impellers of this pump is made up of cycloidal curves, which construction eliminates pockets and compression spaces. There is no metallic contact between the impellers or casing, as a slight clearance is allowed. The arrows indicate the direction of water flow.

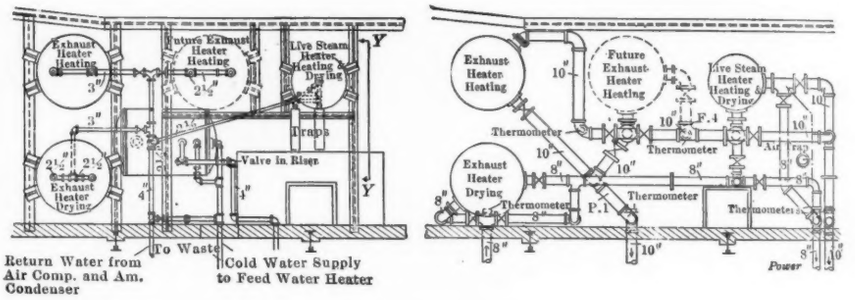


FIG. 12. TOP AND BOTTOM PLAN VIEWS OF HOT-WATER HEATING SYSTEM

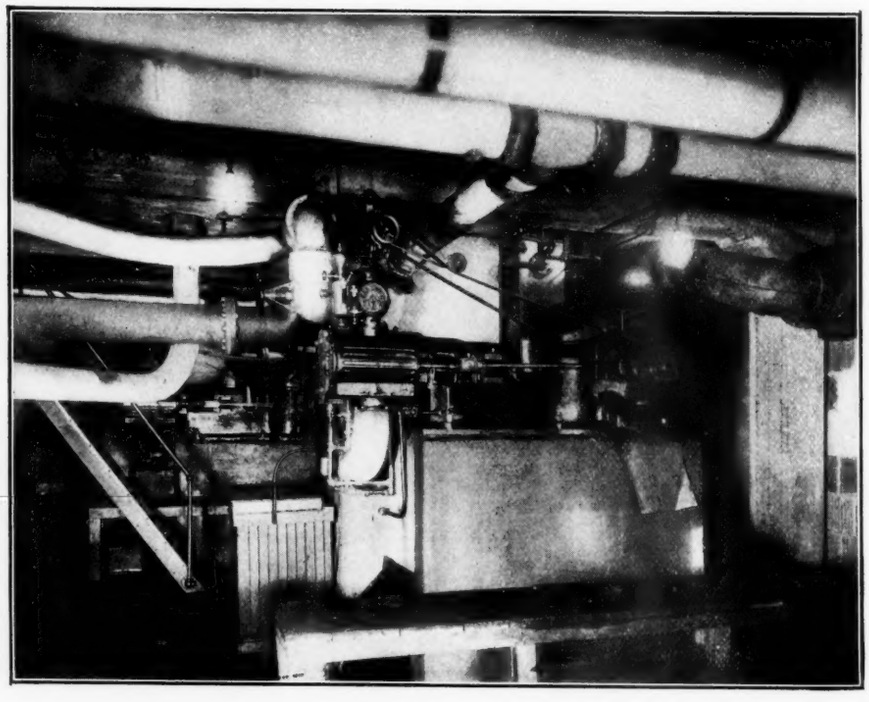


FIG. 10. ENGINE DRIVING VACUUM PUMPS

The water discharged from the vacuum pumps goes to a hotwell from which it is taken by a motor-driven centrifugal pump and delivered to a C. H. Wheeler cooling tower located in the yard. The circulating water flows from the cooling tower to a large cold well, and from the reservoir to the condensers by suction.

The water-cooling tower fans are operated by a motor, but a steam engine, located in the basement, is held as a reserve unit. A large fire pump is also installed in the basement which in case of fire to the factory buildings would doubtless do efficient work.

HOT-WATER HEATING SYSTEM

The hot-water exhaust-heating system was designed by Evans, Almirall & Co. Fig. 13 shows a side and end view of the two systems of heaters and their pipe connections. Fig. 12 shows a top and bottom plan view. There are two separate heating systems which are connected to a live-steam heater, as shown. The Hoppes feed-water heater is placed below the exhaust-steam heaters and the exhaust steam for the high-pressure De Laval turbine, and the steam auxiliaries

escape to the atmosphere through the exhaust header, or can be bypassed to the exhaust-steam heater, as shown in the end elevation of Fig. 13.

turn pipe. As the head of water in both the suction and delivery pipes is balanced, the work of the pump is confined to overcoming friction in the pipes. The

case the exhaust steam does not heat the water to the temperature desired, live steam can be used. The live and the exhaust steam are never mixed, the two

POWER PLANT EQUIPMENT.

Building		Brick.
Boiler room	100 feet wide x 130 feet long	One firing aisle.
Generating room	100 feet wide x 118 feet long	Separated from boiler room by brick wall—concrete floor.
Boilers	12 B. & W	Final total.
At present installed	Six	In two rows.
Size	5000 square feet heating surface	Three steam drums.
Pressure	175 pounds	Saturated steam.
Superheat		None.
Grates	Mechanically operated	Murphy stokers.
Steam piping	6-inch manifold, 15-inch header	8″ expansion bend placed horizontally over boilers.
Feed-water heater (1)	Open type	Hoppes
Boiler-feed pump (1)	14 and 9 by 18 inch	Horizontal simplex, Cameron.
" " " (1)	2½-inch, 4-stage	D'Olier centrifugal, turbine driven.
Hot-water heating system		
Pumps (3)	Centrifugal De Laval	De Laval turbine driven.
Hotwell pump	Centrifugal, D'Olier, 10″	Synchronous motor driven.
Hydraulic pump (1)	21 and 3½ by 24 inch	Knowles duplex.
" " (1)	19, 30 and 4½ by 24 inch	Worthington comp. duplex.
Fire pump (1)	20 and 12 by 16 inch	Knowles duplex.
Ammonia-condenser pump (1)	6 and 7½ by 6 inch	Worthington duplex.
Coal and ash handling		
Conveyer	Link-Belt bucket type	15-horsepower motor.
Capacity	30 tons per hour	Crusher motor, 15 horsepower.
Bunker (1)	Six compartments	Total capacity 1500 tons.
Foundations	Concrete	Turbines, pumps, etc.
Turbo-generators (1)	1000 kilowatts	Allis-Chalmers.
" " (1)	500 kilowatts	Allis-Chalmers.
" " (1)	200 kilowatts	De Laval (two 200-kilowatt generators).
Speed	1800, 3600, 10,500 r.p.m. respectively	De Laval generator, 900 r.p.m.
Generators	3-phase, 60-cycle, 480 volts	Bullock.
Condenser (2)	Jet, Allis-Chalmers	Rotary cycloidal vacuum pump.
Auxiliary turbines		
Hot-water circulating pump turbine (2)	4400 r.p.m	De Laval turbine.
" " " " " " (1)	1700 r.p.m	Motor driven.
Turbine-driving boiler-feed pump (1)	2500 r.p.m	Terry turbine.
Exciters		
Motor driven (1)	D. C., 50 kilowatts 125 volts	Used as a general exciter unit, Crocker-Wheeler.
Steam driven (1)	D. C., 27 kilowatts, 125 volts	Emergency unit, General Electric.
Steam engines (1)	10x12 inch, speed 305 r.p.m	
" " (1)	11x16 inch, speed 90 r.p.m	Fan engine.
" " (2)	8x10 inch, speed 97 r.p.m	Vacuum pump.
" " (1)	12x12 inch, speed 80 r.p.m	"
Motor (1)	60 cycle, 3-phase	Crocker-Wheeler driving exciter.
" (1)	50 horsepower, 3-phase, 60-cycle, 850 r.p.m	Bullock driving air compressor.
" (1)	3 horsepower, 1800 r.p.m	Operates economizer.
" (1)	6 horsepower, 1130 r.p.m	" " "
" (1)	15 horsepower, 680 r.p.m	Operates coal conveyer.
" (1)	15 horsepower, 680 r.p.m	Operates coal crusher.
Ammonia compressor (2)	Steam cylinder 11½x15 inch Ammonia cylinder 11x15 inch	York, 30 tons each.
Air compressor (1)	12x14 inch, speed 115 r.p.m	Motor driven.
" compressor (1)	Steam cylinder 11 and 18 by 14 inch Air cylinder 12x14 inches 125 r.p.m	Steam driven.
Crane (1)	25-ton electric	Niles.
" (1)	10-ton hand	"

The water is circulated through a series of pipes in each heater by a turbine-driven centrifugal pump. From the heat-

water is circulated over and over again, and the loss is slight.

Exhaust steam is passed through the

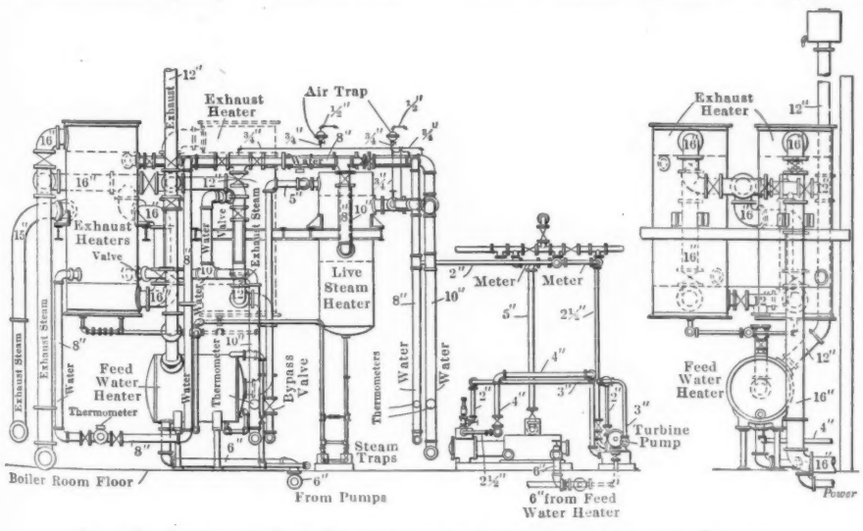


FIG. 13. SIDE AND END ELEVATION OF HOT-WATER HEATING SYSTEM

ers the water is sent to the factory radiators through a single supply pipe and is returned to either of the three centrifugal pumps through a single re-

tubes of the heaters and the water is circulated on the outside.

Besides the two exhaust-steam heaters there is a live-steam heater, so that in

heaters being entirely separate. This permits of returning the live steam to the boiler without passing through a purifying device. The piping is so arranged that the water can be sent through either or both heaters. The water first passes through the Hoppes heater to the bottom exhaust heaters, and then passes out at the top to the heating system. The exhaust steam from the auxiliary engines first passes through either or both of the exhaust heaters to the Hoppes heater and then to the exhaust pipe and the atmosphere. Bypasses are arranged so that either the feed-water heater or the exhaust heaters may be cut out of service.

All steam, water and air pipes are placed in the basement. The main steam pipe is 15 inches in diameter and is fitted with van-stone joints. Long-radius bends are used on all live-steam piping throughout the plant. The table gives details regarding the size, type and use of the various machinery installed in the plant.

The operating force of this plant consists of ten men, which includes both day and night shifts.

Water Rates for Simple Engines

By Wallace S. Fowler

By using the accompanying curve as a basis the water rates for various sizes of simple engines, both condensing and noncondensing, may be figured for different steam pressures and conditions of load.

While connected with a well known concern manufacturing small steam engines, the writer was assigned the task of procuring and tabulating data relative to the power, efficiency and water consumption of the various sizes of engines operating under different conditions of load and steam pressure, both condensing and noncondensing.

Accordingly, complete data were obtained covering sizes from 5x5 to 16x16 inches, with steam pressures ranging from 40 to 200 pounds and loads varying from 25 per cent. normal to 25 per cent. overload. The engines were all of the ordinary piston or slide-valve type, direct connected to generators, and full load was based at one-third cutoff. The problem now was to condense this mass of data into such form as would be readily available to the salesmen and engineers who were to use it. A 14x14-inch engine running noncondensing on 100 pounds steam pressure was chosen as a basis for constructing the fundamental curve shown in the cut, from which the performance of the other sizes might be figured. The water rate, expressed in pounds of water per indicated horsepower per hour, was spaced off as ordinates; the load, expressed in fractions of the normal load, was laid off as abscissas, and a smooth curve was drawn through the points of intersection. This done, cards showing the power, efficiency and water consumption of the standard sets, were made similar to the form shown in the table. These, together with the curve for figuring odd sizes, made a ready-reference library, which was convenient and reasonably correct.

Perhaps the greatest merit of a curve of this kind lies in the aid it renders to the rapid comparison in economy of an engine under different conditions, or of different engines under the same or different conditions. A practical example will serve best to illustrate this point. Suppose that a prospective purchaser requires an engine that will deliver 38 brake horsepower for ten hours per day. He has two engines under consideration, one a 7x7-inch running at 375 revolutions per minute, to be run at one-third cutoff on a steam pressure of 150 pounds gage; the other, a 9x9-inch running 300 revolutions per minute, also to be run at one-third cutoff, but on 90 pounds gage. Both are noncondensing, and either will deliver 38 horsepower. The curve will help him to decide by showing which engine will do the work with the least expenditure of steam. First, let it be determined how much steam the 7x7-inch engine will use. Starting with the point marked 32.3, which is the water rate at full load, and following carefully the rules given below the curve, the problem can be expressed as follows:

$$32.3 + (7 \times 0.03) - (5 \times \tfrac{1}{2}) = 36.6$$

pounds of water per indicated horsepower per hour. Assuming a mechanical efficiency of 90 per cent., this equals,

$$\frac{100 \times 36.6}{90} = 40.7$$

pounds of water per brake horsepower per hour.

Proceeding in the same manner for the 9x9-inch engine,

$$32.3 + (5 \times 0.03) + (1 \times 1) = 38.2$$

pounds of water per indicated horsepower per hour, and, as in the other case with a mechanical efficiency of 90 per cent., it equals,

$$\frac{100 \times 38.2}{90} = 42.5$$

pounds of water per indicated horsepower per hour.

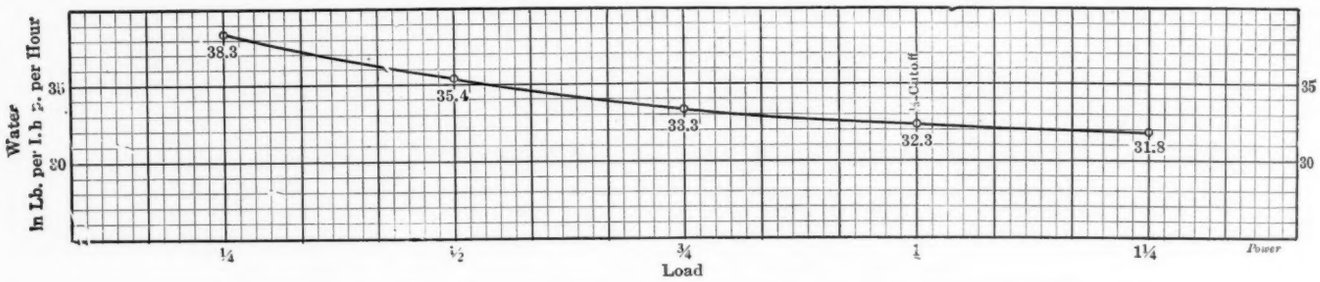
Having thus determined the water consumption, the corresponding coal consumption can be found by substituting the proper values in the following expression:

$$C = \frac{W \times H}{V \times E}$$

where

C = Coal burned per brake horsepower per hour;

W = Water used per brake horsepower per hour;



CURVE OF WATER CONSUMPTION. SIMPLE ENGINES AT DIFFERENT LOADS, CONDENSING AND NONCONDENSING

The curve shows the water rate for a 14x14-inch engine running noncondensing on 100 pounds steam pressure. For condensing engines subtract 15 per cent. For smaller sizes add 3 per cent. for each inch decrease in the stroke. For larger sizes subtract 2½ per cent. for each inch increase in the stroke. If steam is above 100 pounds and noncondensing subtract ½ pound for each 10-pound increase. If steam is below 100 pounds and noncondensing add 1 pound for each 10-pound decrease. If steam is above 100 pounds and condensing subtract ¼ pound for each 10-pound increase. If steam is below 100 pounds and condensing add ½ pound for each 10-pound decrease.

WATER CONSUMPTION OF 14x14-INCH SIMPLE ENGINE.

120 Pounds Steam Pressure, Non-Condensing, 250 Revolutions per Minute, 100 Kilowatt Generator.

Load.	Output in Kilowatts.	Output in Effective Horsepower.	Brake Horsepower of Engine.	Indicated Horsepower of Engine.	Efficiency of Generator.	Efficiency of Engine.	Combined Efficiency.	Water per Kilowatt.	Water per Effective Horse-power-hour.	Water per Indicated Horse-power-hour.	Water per Brake Horse-power-hour.
1¼	125	167.5	185	196.2	90.5	94.3	85.4	48.3	36.2	30.8	32.6
1	100	134	149	160.2	90	93	83.7	50.2	37.4	31.3	33.7
¾	75	100.5	113	124.2	89	91	81	53.5	39.5	32.3	35.5
½	50	67	77	88.2	87	87.4	76	60.7	45.3	34.4	39.4
¼	25	33.5	42.7	53.9	78.5	79.2	62.2	80.5	60.1	37.3	47.2

H = Total heat in the steam above the temperature of the feed water;
V = Heating value of the coal;
E = Efficiency of the boiler.

In the foregoing problem assume that
V = 13,000 B.t.u.;
E = 65 per cent; and that the temperature of the feed water is 32 degrees Fahrenheit.

Then from the steam tables H is found to be 1193.5 B.t.u. for 150 pounds gage and 1182.9 B.t.u. for 90 pounds gage. Substituting these values for the 7x7-inch engine,

$$C = \frac{40.7 \times 1193.5}{13{,}000 \times 0.65} = 5.75$$

pounds of coal per brake horsepower per hour, and for the 9x9-inch engine,

$$C = \frac{42.5 \times 1182.9}{13{,}000 \times 0.65} = 5.95$$

pounds of coal per brake horsepower per hour.

By this it is shown that if the purchaser were to install the higher pressure engine his coal bill would be 3.36 per cent. less than it would be with the other engine. Having found the coal burned per brake horsepower per hour, it is a simple matter to compute the coal bill for the month or the entire year for that matter.

Problem in Expansion of Steam

By John French

Given the initial pressure in an engine cylinder and the ratio of expansion, an approximate solution for determining the pressure at the end of the stroke is presented.

There have appeared at various times articles containing problems involving the expansion of steam. Most of the writers seem to have a rather hazy idea of just how to solve such problems.

The accurate solution is rather long and tedious, but there are approximate solutions which are fairly accurate.

Assume a problem in which dry saturated steam enters a cylinder at a pressure of 100 pounds gage and it is required to find the pressure of the steam at the end of the stroke, with an expansion ratio of three. If a temperature-entropy diagram is drawn, as shown in Fig. 1, it is a graphical history of the steam, where $A\,C$ is the water line and $D\,F$ the saturated steam line. If the steam could expand along the line $D\,F$, which it cannot do unless heat is added, it would be a very easy matter to find the

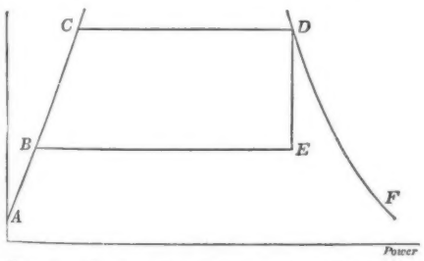


FIG. 1. TEMPERATURE-ENTROPY DIAGRAM

pressure for any volume direct from the steam tables. The most probable expansion line will be approximately an adiabatic such as appears in the ordinary treatment of the Rankine and Carnot cycles and is shown as $D\,E$ in Fig. 1, the complete Rankine or Carnot cycle being $B\,C\,D\,E\,B$.

A crude approximate solution is obtained by assuming the law

$$p\,v = Constant,$$

and a more accurate approximate solution would be given by assuming

$$p v^{1.11} = Constant$$

If the problem given above is solved by the equation

$$p\,v = Constant,$$

then,

$$p_1 v_1 = p_2 v_2$$
$$115 \times 1 = p_2 \times 3$$
$$p_2 = \frac{115}{3} = 38.3 \text{ pounds absolute}$$

or 23.3 pounds gage, assuming for the sake of simplicity atmospheric pressure to be 15 pounds. Using

$$p\,v^{1.11} = Constant$$

the solution is

$$p_1 v_1^{1.11} = p_2 v_2^{1.11}$$
$$115 \times 1 = p_2 \times 3^{1.11}$$

and

$$p_2 = \frac{115}{3.389} = 33.9 \text{ pounds absolute}$$

or 18.9 pounds gage.

If an exact solution is made by the plotting of pressure and volume for constant entropy, the procedure is as follows:

From the result found by using the equation

$$p\,v^{1.11} = Constant$$

it is known that the pressure will be somewhere in the neighborhood of 34 pounds absolute. Assuming the steam to have expanded adiabatically until the pressure is 34 pounds, the first factor to be found is the quality at this pressure. Then the volume is found and if it is equal to three times the volume of the steam at the initial pressure, the required pressure has been ascertained. Using Reeve's steam tables it will be found that the volume of one pound of steam at 115 pounds absolute is 3.864 cubic feet so that the final volume will be 11.592 cubic feet.

The entropy of dry saturated steam at 115 pounds absolute is equal to the entropy of the water plus the entropy of vaporization for that pressure and the entropy of dry saturated steam at 34 pounds is equal to the entropy of the water plus the entropy of vaporization for that pressure. But at the final pressure the steam is not dry, so the entropy will be equal to the entropy of the water plus the entropy of vaporization multiplied by the quality.

Let E_1 and E_2 be the entropy for 115 and 34 pounds, and x the quality. Since the entropy at both pressures is equal,

$$E_{w_1} + E_{v_1} = E_{w_2} + x E_{v_2}$$
$$x = \frac{E_{w_1} + E_{v_1} - E_{w_2}}{E_{v_2}}$$

and

$$x = \frac{0.4864 + 1.0969 - 0.3784}{1.2999} = 92.66$$

The relative volume of dry saturated steam at 34 pounds is 12.11, so that the volume of the steam with a quality of 92.66 is 11.22 cubic feet. This is less than the volume required; hence it is necessary to repeat the same process for 33,

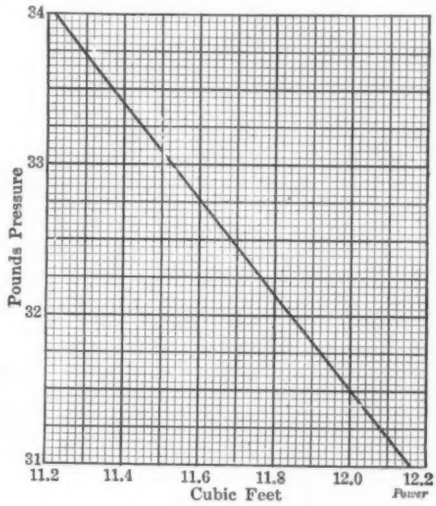


FIG. 2. PRESSURE-VOLUME CHART

32 and 31 pounds. This can most easily be done by arranging a table as follows:

Pressure.	Volume.	E_w	E_v	x	xV
34	12.11	0.3784	1.2999	92.66	11.22
33	12.46	0.3759	1.3048	92.55	11.53
32	12.82	0.3734	1.3098	92.37	11.84
31	13.20	0.3709	1.3149	92.19	12.17

With the values of $x\,V$ as abscissas and those of pressure as ordinates, the chart shown in Fig. 2 can be plotted. At a volume of 11.59 the pressure 32.8 pounds absolute or 17.8 pounds gage will be found, which is the exact pressure of the steam after expanding under the conditions stated in the assumption.

Operating vs. Consulting Engineer

By M. Kennett

An instance where the consulting engineer departed from the custom of heeding the suggestions of the chief, with the result that the layout proved so unsatisfactory as to require complete rearrangement of the equipment involved.

The majority of operating engineers are inclined to feel more or less jealous and resentful toward the consulting engineer who is called in to plan alterations or additions to their plants. This feeling is usually unjustified, as the consulting engineer is undoubtedly better qualified for this work than the operating engineer. Furthermore, my experience has been that the consulting engineer is usually ready to consider suggestions from the operating engineer, and in this way, by combining the technical knowledge of the former with the practical experience of the latter, a much more efficient layout is obtained than would be possible by relying upon either one alone. It was my misfortune, however, to come in contact with a striking exception to such a consulting engineer in connection with the installation of a new pump in an electric-power and pumping plant, in which I was employed some years ago.

This plant was built on the bank of a river, about 35 feet above the low-water level and some 400 feet back from the river; the overflow during periods of high water making it impractical to build the plant closer to the stream. The engine and boiler rooms were built on the surface, but owing to the elevation, the pumps were placed in an oval pit about 15 feet deep and located just in front of the boiler room, as shown in Fig. 1. In this pit were placed two compound direct-acting pumps, one of 1,000,000, and the other of 750,000 gallons capacity. During most of the year these pumps worked on a suction lift of from 10 to 12 feet, but during low water this was increased to about 20 feet in addition to the horizontal pull of about 400 feet, and at times considerable difficulty was experienced in operating them at this lift. Hence, when the increase in business demanded an additional pump, it was decided to place it about 5 feet lower than the old ones. As there was no room in the old pit, it was necessary to sink a new one just in front of it and connected to it by a flight of narrow steps; see Fig. 1.

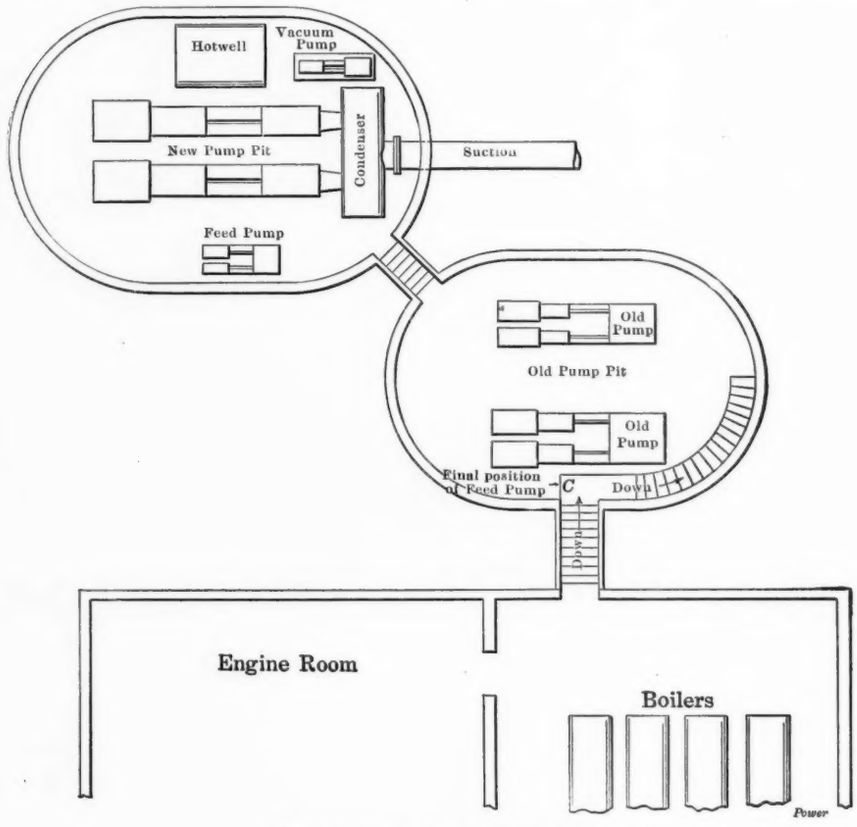


FIG. 1. LAYOUT OF PUMP PITS

Although this new pit was directly in front of the engine room, no stairway from there was provided, as the consulting engineer claimed that it would be necessary only to go down to the new pump room two or three times a day. As will be seen from the plan, it could be reached only by a long route through the old pit, and the new pump could not be seen at all without going to the entrance of the new pit. When the engines were running, it was impossible even to hear the pump, and the only means of judging what was going on was by means of the gages in the engine room.

The new pump was of the compound, condensing type, having steam-jacketed cylinders and a surface condenser in the suction line, so that the entire volume of water flowing through the suction constituted the circulating water. The water end was of the outside-packed plunger type, and the pump was rated at 1,500,000 gallons capacity.

The vacuum pump was placed at the end of the condenser, and in order that the water might flow to it by gravity, it was located in a small pit, about 4 feet long, and 18 inches wide. This vacuum pump discharged into a hotwell which consisted of a cylindrical tank placed on its side on the floor. By means of a float valve the makeup water for the boilers was supplied to this hotwell, from which the feed for the entire plant was taken. The supposition was that when only this pump was running, the water in the hotwell would be too hot to be lifted by suction; consequently, the feed pump was located on the floor of the pump pit, on the side opposite the vacuum pump. This was the layout as planned by the consulting engineer. The chief engineer of the plant, who was a very capable man, found several objections to it, but was unsuccessful in his efforts to have the plans altered, and the installation was completed as outlined.

Some of the objections which the chief raised were that the feed pump was in a very inconvenient place, where it could not be seen by the fireman without going down two flights of steps. Although it could be regulated from the boiler room by a valve in the steam line, it is always more satisfactory to see a pump in order to note the changes in its speed while regulating it. Another objection was that all the seepage, which was considerable, together with the drips from the plungers and stuffing boxes, would naturally flow into the vacuum-pump pit which did not have any drainage and really acted as a sump, and as a result had to be emptied by a jet provided for the purpose. This pit usually filled in about two hours, and the grit soon cut the links and rods of the vacuum pump, and its cramped position made it very difficult to pack or repair the pump.

These were some of the most serious objections to the proposed plan, and to

overcome them to some extent, the chief suggested getting the hotwell out of the way by elevating it about 6 feet, and placing the feed pump on the landing at the bottom of the first flight of steps leading from the boiler room, as at *C*, Fig. 1. This would have located it in plain view from the boiler room, and given it a suction lift of only 5 feet. Furthermore, the chief did not believe the water in the hotwell would ever become hot enough to cause any trouble on this lift.

These suggestions were promptly vetoed by the consulting engineer on the ground that to elevate the hotwell would cause the vacuum pump to operate against a head, thus imposing additional work on it, which might lower the total duty of the pump to less than the guarantee.

To overcome this argument, the chief proposed disconnecting the discharge pipe during the test and letting the vacuum pump discharge on to the floor. But in spite of all arguments to the contrary, the consulting engineer ruled that the hotwell must remain on the floor. Nor could he see any objections to the feed pump being located in the lower pit, and contended that it would not handle the water from the hotwell on a 5-foot lift. He also considered it unnecessary to provide stairs from the engine room to the new pump pit, although their desirability would seem to be apparent to anyone.

Finally, after many tribulations, the installation was completed and the consulting engineer took leave, much to the relief of all. After a few weeks, when things had settled down to regular operating conditions once more, the chief quietly set to work to rearrange things. First, he had the hotwell raised about 6 feet, and suspended from the overhead beams, and then placed the feed pump where he had proposed, on the landing at *C*, which proved to be a much more convenient location; and I have never heard of its giving the least trouble in lifting the water.

The cylinders of the main pump were steam jacketed, and as originally installed, were drained by means of a trap. The chief left the trap in position, but cut it out of service by means of a valve, and took steam for the vacuum pump through the jackets. This kept the steam in better circulation, and kept the jackets thoroughly drained at all times, and the condensation was never enough to affect the operation of the vacuum pump.

An amusing incident occurred when the duty test was made on this installation. The manufacturers, of course, had their representative on hand to look after their interests and the consulting engineer represented the owner. In order to ascertain the total head against which the pump was operating, that is, the hight from the surface of the river to the surface of the water in the standpipe, a vacuum gage was placed on the suction pipe at *A*, Fig. 2. As each inch of vacuum is equal to 0.49 pound, the total pressure represented by the reading of this vacuum gage was added to that shown by the pressure gage, and this total divided by 0.433 gave the total head in feet against which the pump was working. As provision had been made at *B* for a gage, the consulting engineer placed one at that point, but the builder's man objected to this, claiming that the vertical hight between these two gages should be included in the total head. In other words, as this distance was about 6 feet

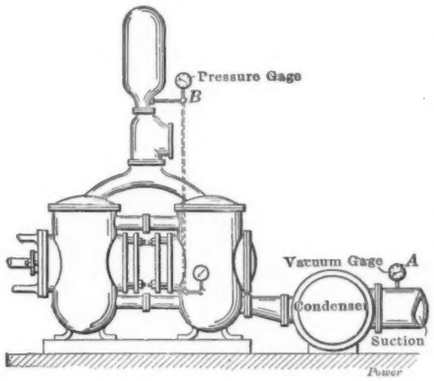


FIG. 2. CIRCULATING PUMP SHOWING LOCATION OF GAGES

he claimed that for every pound of water raised, the pump was developing 6 foot-pounds of energy for which it was not receiving credit, and he insisted that the pressure gage be located on the same level as the vacuum gage, as shown by the dotted lines in Fig. 2.

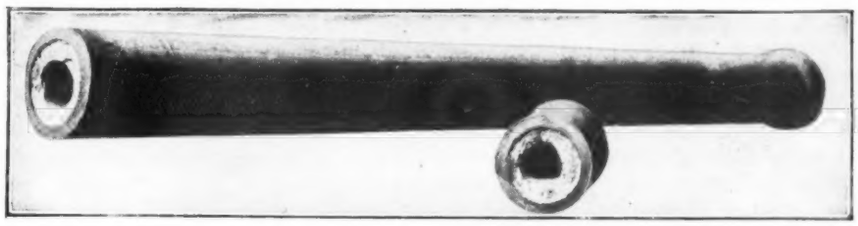
After considerable argument on this point it was evident that they could not agree, and each made the test with the gages where he thought proper, and the question was left to be decided later by a committee of consulting engineers. These men decided in favor of the builders, and under this decision the pump exceeded the guarantee by a considerable amount.

This test was more or less perfunctory, and no attempt was made to attain any great degree of accuracy. The pump was guaranteed to develop 55,000,000 foot-pounds of energy per 1000 pounds of steam at 100 pounds pressure at the throttle. However, no test was made to ascertain the quality of the steam, which must have been rather poor, as the pipe was about 100 feet long and no separator was provided, although the pipe was well protected by nonconducting covering. The quantity of water pumped was calculated by the displacement of the plungers, and the number of strokes as indicated by a counter, the length of the stroke being frequently checked during the run. An allowance of 5 per cent. was agreed upon for slip through the valves and packing. The discharge from the vacuum pump and from the trap which drained the cylinder jackets was carefully weighed and represented the total steam consumption. The head against which the pump was working was computed as described from the combined readings of the pressure gage on the discharge and the vacuum gage on the suction line. A short run of four or five hours was made, the consulting engineer and the representative of the builders each keeping their own notes. Naturally, there was quite a difference between the two results, which, as stated above, was finally decided in favor of the builders.

During the daylight run, the electrical load was very light, and considerable trouble was experienced in keeping the feed pump running slowly enough to maintain a constant water level in the boilers. When it was throttled down so as to run slowly enough, it frequently stopped altogether. A ¼-inch bypass with a globe valve in it was inserted between the discharge and the suction and the pump was then run at a conveniently slow speed, and the bypass was regulated according to the water level in the boilers. This was found more convenient than attempting to regulate the pump too closely.

Failure of Scaled Blowoff Pipe

One of the reasons a blowoff pipe burns out is because it is allowed to become partly filled with scale. The illustration shows a 2½-inch blowoff pipe which failed from this cause. It can readily be seen that with such an accumulation of scale, the pipe had but little protection against the furnace heat.



SCALE IN 2½-INCH BLOWOFF PIPE

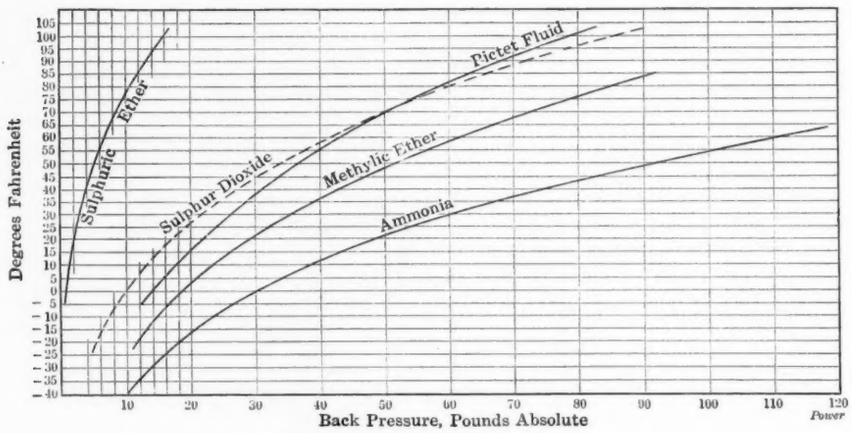
The pipe was lined with scale between the boiler and the blowoff pipe. The scale was of lime formation and exceedingly hard. A peculiar feature was that the scale seemed to vary in character, an end view showing different colored scale of varying thickness; the layers being easily distinguished. A blowoff pipe should be inspected just as frequently and as thoroughly as a boiler.

Pressures in Refrigerating System

By F. E. Matthews

The capacity of a refrigerating system may be increased by lowering the back pressure or by adding to the cooling surface. Some interesting data are given on temperatures and pressures and their relations.

It has already been pointed out that a given substance boils at different temperatures under different pressures; the boiling point being raised when the pressure is increased, and lowered when it is decreased. In the case of water, for example, which boils under atmospheric pressure at 212 degrees Fahrenheit, an increase in pressure to 70 pounds gage raises the boiling point to 316 degrees Fahrenheit, and a reduction in pressure to 29.74 inches vacuum lowers it to 32 degrees Fahrenheit, or to its freezing point. From this, since the law is a general one applying to all known liquefiable gases, it follows that to produce low temperatures the pressure on the refrigerating medium employed must be reduced to such a point that the corresponding boiling point will be a sufficient number of degrees below the temperatures to be produced to bring about the heat transfer through the expansion coils or other cooling surfaces. If, for example, it is desired to cool a cold-storage compartment at 10 degrees Fahrenheit, a back pressure of 24 pounds gage will be found too high to allow ammonia to boil at this temperature. At 23.64 pounds pressure it will boil at exactly 10 degrees Fahrenheit, but since this is the temperature of the surrounding air, there is no difference in temperature to bring about a heat flow and the boiling will not continue. When the pressure is reduced to 19.46 pounds gage, the ammonia will boil at 5 degrees Fahrenheit, and at this temperature there will be sufficient inflow of heat from the 10-degree surrounding air to cause quite appreciable refrigeration. A further reduction to 15.67 pounds gage lowers the temperature of the boiling ammonia to zero degree Fahrenheit and the increase in temperature difference from 5 to 10 degrees Fahrenheit will effect a rate of heat transfer just twice as great per square foot of pipe surface as was possible with half the difference. A still further reduction to 9.1 pounds gage pressure will allow the ammonia to boil at 10 degrees Fahrenheit, at which temperature the heat flow from the 10-degree room will be twice as great as it was at 15.67 pounds pressure and four times as great as it was at 19.46 pounds. In order to produce the same amount of cooling effect at 19.46 pounds pressure as was obtained at 9.1 pounds pressure, just four times as much pipe surface would have to be employed, and in order to do as much at 15.67 pounds, just twice the surface would be required.



VAPOR TENSIONS FOR VARIOUS MEDIA

If, instead of direct expansion, brine circulation is employed, it will be evident that for the same rate of heat flow, a lower pressure and temperature will be required in the latter case. Assuming, for example, that the rate of heat transmission per square foot per degree difference in temperature be the same between the ammonia and brine, and between the brine and air as it is between the ammonia and air (an assumption which is not wholly accurate but which will simplify the example), the heat transmission between the ammonia at 9.1 pounds pressure and the air at 10 degrees Fahrenheit would be only half as great in the case of the brine system as in the case of direct expansion. This because of the fact that 10 degrees difference in temperature must be allowed to cause a heat flow from the air to the brine and another 10 degrees for the flow from the brine to the ammonia. On this basis in order to produce the same heat flow, 9.1 pounds back pressure would have to be carried in the case of brine circulation against 15.67 pounds in the case of direct expansion.

Where the piping is installed and cannot be increased, there remains only one of the two variables. To increase its cooling capacity, therefore, lower back pressures must be employed.

The saving in first cost by the installation of scanty pipe surface always entails correspondingly lower back pressures, and is soon lost by increased operating expense due to decreased efficiency.

Where there is only one temperature to be produced in the cold-storage compartments a back pressure is usually carried such that the temperature corresponding to that pressure will be from 10 degrees or less on low-temperature work to 30 degrees or more on high-temperature work, below that of the cooler temperature. The lower the temperature to be produced, the lower the efficiency of the machine, and accordingly the more expense will be warranted in pipe area so as to increase transmission at a narrower range. Under average operating conditions the cost of the amount of expansion pipe required to allow this range in temperatures balances up fairly well with the loss in efficiency that would be encountered if less expansion piping were installed, and a lower back pressure carried.

Where several different temperatures are to be maintained with one back pressure, no fixed rule can be followed and each individual case must be figured out separately. If only a small percentage of the total cooling be low-temperature work, it is usually advisable to increase the surface and to reduce the temperature range between the liquid ammonia and the surrounding air in the lowest temperature compartment. In this case the use of an abnormal amount of pipe on a small percentage of the entire duty tends to increase the efficiency of the whole plant.

Some idea regarding the pressures that should be maintained in expansion coils when operating with different kinds of refrigerating media may be gained by reference to the curves, from which it may be seen that if an expansion temperature of zero degree Fahrenheit, such as would ordinarily be employed where the temperatures to be produced are from 10 to 20 degrees Fahrenheit, the expansion or "back pressures" required for these different refrigerating media are as follows: Ammonia, 30 pounds absolute; methyl ether, 19; pictet fluid, 14; sulphur dioxide, 10.

While no definite rules can be laid

down regarding the back pressures that should be carried even under average conditions, the pressures and temperatures given in Table 1 will be found to be fairly accurate.

TABLE 1. BACK PRESSURES AND TEMPERATURES.

Temperature of room, degrees Fahrenheit	5	10	15	20	28	32	36	40	50	60
Back pressure, pounds gage	7	10	12	15	22	25	27	30	35	40
Temperature of ammonia, degrees Fahrenheit	13	10	5	0	8	12	4	17	22	26

In general the engineer should endeavor to so manipulate his expansion valves as to carry the highest back pressure possible and still produce sufficient refrigeration in his coldest coolers. A second limit to possibilities in this direction is reached when no more ammonia feed can be put on the expansion coils without causing liquid ammonia to return to the compressor, causing it to pound and the rod stuffing boxes to leak. It may be mentioned incidentally that the back pressure can be carried materially higher, and the efficiency of the plant materially increased by keeping the expansion coils free from the insulating effect of ice on the outside and oil on the inside. The coating of ice should be kept as thin as possible at all times and opportunity should be taken to remove oil from the expansion coils once in every two or three seasons at most, and oftener if much oil is thought to have worked into the system.

When the temperatures get too low, it is better to slow down the machine rather than to reduce the refrigerating effect of the system by closing down expansion valves, reducing the back pressure and literally throwing away power because the refrigeration is not needed. The full importance of this truth is seldom recognized by either the supervising or operating engineer, and it is rarely that either strives for the last pound of increased back pressure half as diligently as for the last inch of vacuum in the steam condensers, although the former pressure is of far more importance than the latter in its effect on the general efficiency of the plant.

Condenser Pressure

Just as the back pressures have to be reduced in order to reduce the boiling point of the refrigerating medium when low temperatures are required, the condenser pressure always rises sufficiently to raise the boiling point of the medium when the temperature of the cooling water is raised. Table 2 gives the approximate condenser pressures that should result from the use of different quantities of cooling water of different temperatures on condensers of average proportions.

In every event the condenser pressure should be kept as low as possible and the back pressure as high as the temperatures to be produced will permit, narrow limits between such pressures being as important to the efficiency of a refrigerating system as wide ones are to that of a steam engine, in which the economy increases with the range between boiler pressure and condenser pressure.

TABLE 2. CONDENSER PRESSURES AND TEMPERATURES.

1 gallon per minute per ton per 24 hours—							
Temperature of cooling water, degrees Fahrenheit	60	65	70	75	80	85	90
Condenser pressure, pounds gage	183	200	220	235	255	280	300
Temperature of condensed liquid ammonia, degrees Fahrenheit	95	100	105	110	115	120	125
2 gallons per minute per ton per 24 hours—							
Condenser pressure, pounds gage	130	153	168	183	200	220	235
Temperature of condensed liquid ammonia, degrees Fahrenheit	77	85	90	93	100	105	110
3 gallons per minute per ton per 24 hours—							
Condenser pressure, pounds gage	125	140	155	170	185	200	215
Temperature of condensed liquid ammonia, degrees Fahrenheit	75	85	90	93	95	100	105

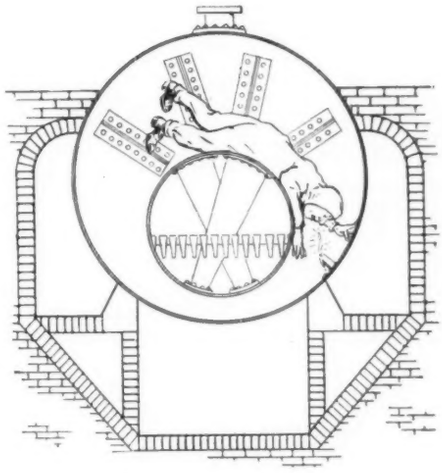
Remarkable Presence of Mind

By G. James Wells

Mr. Clark was a boiler expert, and what he did not know about a steam boiler was not worth anyone's attention. His favorite type was that known as the "Lancashire," and if an acquaintance wished to annoy him he could easily do so by suggesting that any water-tube boiler was superior to the Lancashire type.

Now, Mr. Clark, undoubtedly, was competent to express an opinion upon the merits or demerits of a boiler of any type, for he had spent his whole life around boilers, beginning as a riveter and plater and later becoming a charge hand, and proving so reliable that he was frequently chosen for work away from the shop. When away, he was obliged to send daily reports to the office, and in this way he soon learned to express himself in fair language, and having studied the board of trade reports upon boiler accidents, he was made an inspector.

As may be imagined, he was now on the road to further advancement and upon leaving the shop became a recognized member of the office staff. So far he had had no serious setback, and the indications were that he would prove eminently successful in his new capacity as inspector. His thorough knowledge



of the trade as an expert workman, coupled with his natural shrewdness in detecting signs of deterioration, made an ideal combination for an inspector.

A customer having some boilers of the Lancashire type, one of which an insurance company had reported upon as requiring repairs, appealed to Mr. Clark's firm for advice and an estimate for doing the necessary work. This led to his receiving instructions to make a thorough examination and report at once. Accordingly a start was made for the factory, and upon reaching the boiler room he made a superficial examination, externally, and requested that the flues be thoroughly cleaned. He also marked certain portions of the brickwork for removal which his experience had taught him might reveal wasting of the plates. While this was being done, he entered the boiler and began a systematic examination of every plate, rivet and joint. One of the effects of heat upon boiler plates is to expand them, the hottest portions expanding most; hence deformations are frequently found which are due to the unequal expansion of the shell and flues. Resulting strains are produced, which give rise to bending at the flanges where the end plates are attached to the shell and flues. This movement takes place when the temperature of the boiler is changing, and is termed "breathing," and in time the plates commence to crack. Water entering the crack may cause corrosion and assist in increasing the depth of the crack. For this defect Mr. Clark was naturally on the lookout, and in order to be certain that he had seen all round

each seam of the flues examined, he marked with a piece of chalk the limits of his view from any position. Thus he viewed the under parts of the flue seams by wriggling about on his back beneath the flues, afterward completing his inspection by lying on the top of the flue and reaching over until he could see his mark.

The most convenient light to use on such occasions is the old -fashioned tallow candle, which easily supports itself by adhesion when squeezed against some rough surface. Now it is not always easy to determine whether an apparent crack is really a crack in the plate or only a crack in the deposit upon the plate, and due to a little uncertainty upon this point Mr. Clark allowed himself to slip down head first between the flue and shell, as far as possible, in order to see better. However, he did not notice the position of the candle until suddenly his nose was uncomfortably close to the flame. His position was such that he could not get either hand to shift the candle, for he needed both to support himself from going farther down; so he began shouting for assistance. He yelled loudly and continuously, but to make himself heard and understood from the interior of the boiler, when a candle was burning his nose was not easy. Finally one of the men who heard the calls for help put his head into the manhole opening and asked, "What do you want?"

Mr. Clark, who was at no time very patient, on this occasion lost all that he possessed, and replied in suitable language with a demand that the man pull him back as the candle was burning him. After an instant's reflection the young fellow instead of instantly dropping into the boiler to assist the man inside called out, "Can't you blow out the candle?" The candle was instantly blown out and the crisis ended, after which Mr. Clark began wriggling his way back to the top of the flues, meanwhile wondering why he had not thought of blowing out the candle in the first place, also how it happened that when shouting he had not put out the candle.

Unfortunately, his nose was badly burned and occasioned him a great deal of physical pain, with the additional annoyance of explaining to his friends the origin of the trouble. His friends, although quite sympathetic, could not hide their appreciation of the humorous side of the incident, so he decided to avoid publicity by retiring until his nose had recovered its normal appearance.

Theory of Chimney Draft

By E. J. Kunze

Expansion due to heat, the force of gravity and pressure due to weight of the atmosphere are factors which produce the phenomenon of natural draft.

The term draft is sometimes used to indicate the volume or weight of gases passing through a fire. It would be better, when speaking of draft, to refer to a difference in pressure between two different points in the passage of a gas; but it is more convenient to indicate the difference in pressure between the external air and the gases at the point of measurement and draft is, therefore, usually so considered.

There are at least three explanations given for the phenomenon of chimney draft. It has been said that it was caused by heat, by pressure and by gravitation. Of recent years the latter has been the one most generally accepted. Parsons* says, "The problem of chimney draft is really one of graviation and not one of thermodynamics." While it is true that the action of gravity plays a part in the establishment of draft, it is equally true that there can be no difference in the specific gravity of any flue gases due to heat unless thermodynamic actions take place. It will be seen, therefore, that each of the three above-mentioned agents plays its part.

The tendency of nature is always toward equalization, reducing the abnormal to the normal; this is true of temperatures, of pressures and of forces of any kind. In the case of draft, while temperature is indeed equalized, no current is set up directly by the interchange of heat. The hotter gases lose heat to the colder gases but this does not cause the fluid to move. In the case of hot gases, a very small amount of heat is transmitted by conduction, the major portion being transmitted by convection due to the movement of the gases. Heat transmitted by radiation cannot, directly, set up a current because heat is so transmitted through both a denser and a rarer medium, and even through a vacuum, although it may not be sensed unless some material substance is set in vibration. If this were not so the heat energy of the sun would have been spent long before it reached the earth.

Although a drawing action may be implied in the term draft, there can be no such action in dealing with air or any other gas. A pushing rather than a pulling action exists. The gas moves from a place of higher pressure to one of lower pressure.

According to the kinetic theory of heat, the movements of the molecules of a gas when heated become more violent; hence, due to the mutual blows imparted to one another, the molecules are more and more separated from each other and in this way the density in different parts is changed and the specific gravity is lessened. The vibrations of the molecules do not cause the molecules set in vibration to be transferred from one position to another, the molecule is merely more or less agitated without causing it to lose its approximate position and the vibration, not the molecule, is transmitted along the medium by the action of the heat. But these vibrations cause the molecules affected to be repulsed by each other and hence a decrease in density occurs. The inequality thus set up causes a flow of the liquid or gas from the denser to the rarer portion and the denser gas pushes the rarer gas along until the latter reaches a place where the pressure is equal to its own.

In the case of chimney draft, the gases inside the stack are expanded due to the heat. Hot air weighs less than cooler air under the same pressure. The column of air inside the chimney weighs less, therefore, than an equal column of outside air. The pressure of the gas at the base of the stack will be less than the pressure of the outside air at that point. Suppose the gas were to expand fully before reaching the top of the stack, then the pressure of the gas would be equal to that of the air at the top of the stack. If the lower part of the chimney were to have communication with the outside air at that point, the heated gas would rise because of the difference in pressure of the air at the two ends of the chimney. This follows the law of gravity and for this reason, draft in a vertical flue is better than it would be under otherwise like conditions in a horizontal flue. The tendency is for the pressure to equalize, and it may seem most natural for the pressure of the outside air to be communicated tc the inside of the chimney through the top but this is not the case.

The expansion of a colunm of gas or air due to heat must take place in the direction of least resistance which, in the case of flue gases is, obviously, toward the top of the chimney. The pressure of the gas at the top of the chimney will be less than the pressure of the outside air at that point; first, because the gases have not been fully expanded, as a rule, by the time they have reached the top; secondly, the momentum of the moving gases increases the pressure in the direction of motion. The air, therefore,

*"Steam Boilers," page 125.

could not enter at the top. Hence, in order to tend toward equalization of pressure at the base of the chimney, the air must enter through the grate and cause the phenomenon of draft. Of course, the pressure at the base of the stack is not exactly that of the outside air at that point because of the loss in passing through the bed of fuel and ash. The higher the chimney, other things being equal, the higher will be the velocity of the moving gases and the lower will be the air pressure at the top, hence, the greater the draft.

The explanation of artificial draft follows a similar process of reasoning: With induced draft the pressure between the fire and the stack is reduced by means of a fan or jet of steam, forming a partial vacuum, at the same time a current is set up toward the top of the chimney, the gases being thus forced out by pressure. The air, therefore, must, as before, enter through the fire. In the case of forced draft, the air is forced in through the fire by means of a blower and in this case the combination of pressure and flow readily follows the explanation of draft given.

Therefore, draft is due to difference in pressure but, in the case of natural draft, this difference in pressure is produced by the expansion of the heated gases, the ultimate direction of the motion being vertical because of the action of gravity, the rarer gas being forced up by the denser gas (air) below it.

Engines with a Minimum of Clearance

In our issue of August 30 we referred to an engine at the Brussels Exposition which has 0.75 of 1 per cent. clearance. We are able with the accompanying engravings to show how this is accomplished. The valves instead of being put in at right angles to the axis of the cylinder are set at an angle of about 30 degrees, as are the valves of the Linde and other air compressors where the suppression of the clearance is of so much moment. The end of the cylinder is either rounded or flattened off to meet this conformation, as shown in the engraving, so that the clearance volume is reduced to the very small space between the surfaces of the cylinder and the piston and the volume of the port *A*, Fig. 1, amounting to no more than 0.75 of 1 per cent. in the smaller engines and to ½ of 1 per cent. in the larger.

With the valves set at an angle to the axis of the cylinder similar to air-compressor construction, with the ends of the cylinder rounded off and the piston made to conform, a clearance of 1-2 of 1 per cent. is made possible.

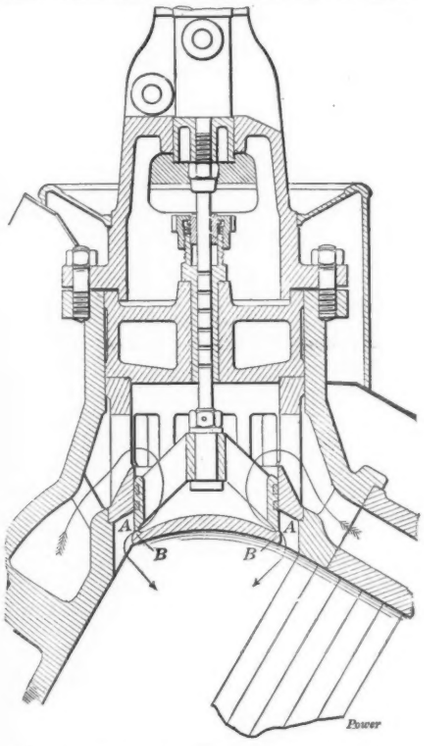


FIG. 1. SINGLE-SEAT VALVE OF WALSCHAERTS' "GIANT" ENGINE

The admission valve is formed of a hollow cylinder with packing rings such as are used for certain piston valves. It does not close the port by passing the edge of it as does a piston valve, however, but seats positively with its lower edge at *B B*, Fig. 1, the steam following the course indicated by the arrows. The covering portion, not having to uncover an admission port, may have a large bearing surface with generously proportioned rings, whereas in the piston valve the parts upon which tightness depends are reduced and limited by the conditions required for opening and closing the port. The rings have a single joint and are not exposed to the irregular wear caused by passing over the bridges or girders of the ports. The difficulties and disadvantages of trying to make two seats simultaneously upon the double poppet are avoided.

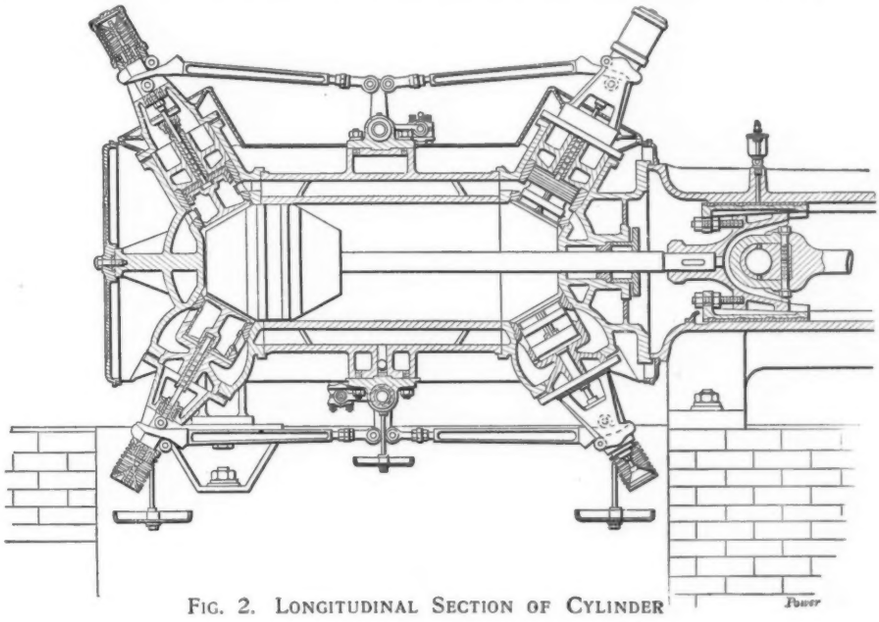


FIG. 2. LONGITUDINAL SECTION OF CYLINDER

The engine is made in this form by Ernest Mennig, successor to Walschaerts, the inventor of the famous valve gear of that name, at St. Gilles, Brussels. He has used single-seated valves for a dozen of years successfully, placing them horizontally above and below the cylinder, but wishing to place them in the cylinder heads was unable with the horizontal arrangement to obtain sufficient area without the use of more than one valve for each position. It was to overcome this difficulty that they were placed in the oblique position shown in the engravings. The valves are well guided, and there appears to be no reason why they should not work as well in this as in either the horizontal or vertical position.

Although the surface initially exposed to the steam is somewhat increased by the hemispherical or conical shape of the piston head and surrounding wall, these surfaces are rendered as harmless as possible by being highly polished and steam jacketed. The admission is controlled by a governor placed on the same shaft as the eccentric and is effected by the simple cam levers shown in the longitudinal view of the cylinder.

Electrical Department

The Parallel Operation of Compound-wound Alternators

By R. L. Mossman

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Compound-wound alternators operated in parallel generally cause more trouble with cross currents than machines that are separately excited only. In one case that I know of this difficulty was so serious that the compound windings had to be cut out of circuit entirely. Two compound-wound alternators were driven by engines having shaft governors that could not be adjusted while the engines were running. The generators were 150-kilowatt revolving armature machines, giving two-phase currents at 2200 volts, and were direct connected to their respective engines. The machines could be put in parallel without any trouble, but when the load came on the cross currents would build up so that one machine would have to be pulled out, and as there were times when one could not carry the load, it was decided to change the winding and make the machines separately excited.

For the benefit of engineers who have not had experience with this type of machine, Fig. 1 is shown; this is a diagram of the circuits of a compound-wound two-phase alternator. The generator field magnet is shown at *G*; *H* represents the auxiliary field winding and *I* the main field winding supplied by the exciter. The armature is shown separately at *J* to make the diagram clearer; *A*, *A'*, *B* and *B'* are the terminals of the two armature circuits, carried to the collector rings *C* and a transformer *K* which is mounted in the armature spider and revolves with it. The secondary winding of the transformer is connected to a two-part rectifying commutator *D*, which changes the alternating current into pulsating direct current in the circuit *N N*. The primary windings of the transformer are in series with the leads *A* and *B*, which are ultimately connected to their respective collector rings at *C*.

The leads *N N* from the brushes of the rectifying commutator *D* connect the auxiliary field winding to the rectifier. The main field winding *I* is connected through the generator field rheostat *R* to the field switch *O*, and thence to the exciter *E*. The type of field switch shown is generally used to protect the field winding when the switch is opened. When a field circuit is suddenly opened there is a momentary inductive kick from the field coils similar to that of an induction coil, the voltage of which is sometimes high enough to puncture the insulation of the field winding. With the kind of switch here shown, when it is opened the winding is short-circuited by the auxiliary blade *P* entering the clips *T*, which are connected to one main blade of the switch through a resistance *S*, thereby preventing the sudden rise of voltage in the field coils.

The compounding effect of the auxiliary field winding in this machine is the same as that of the series winding of a compound-wound direct-current machine. When the load increases, the increased flow of current through the primary windings of the transformer *K* causes an increased flow in the secondary winding and, through the commutator *D*, in the direct current which flows through the leads *N N* to the auxiliary winding, increasing the strength of the field magnet and holding up the voltage. When the

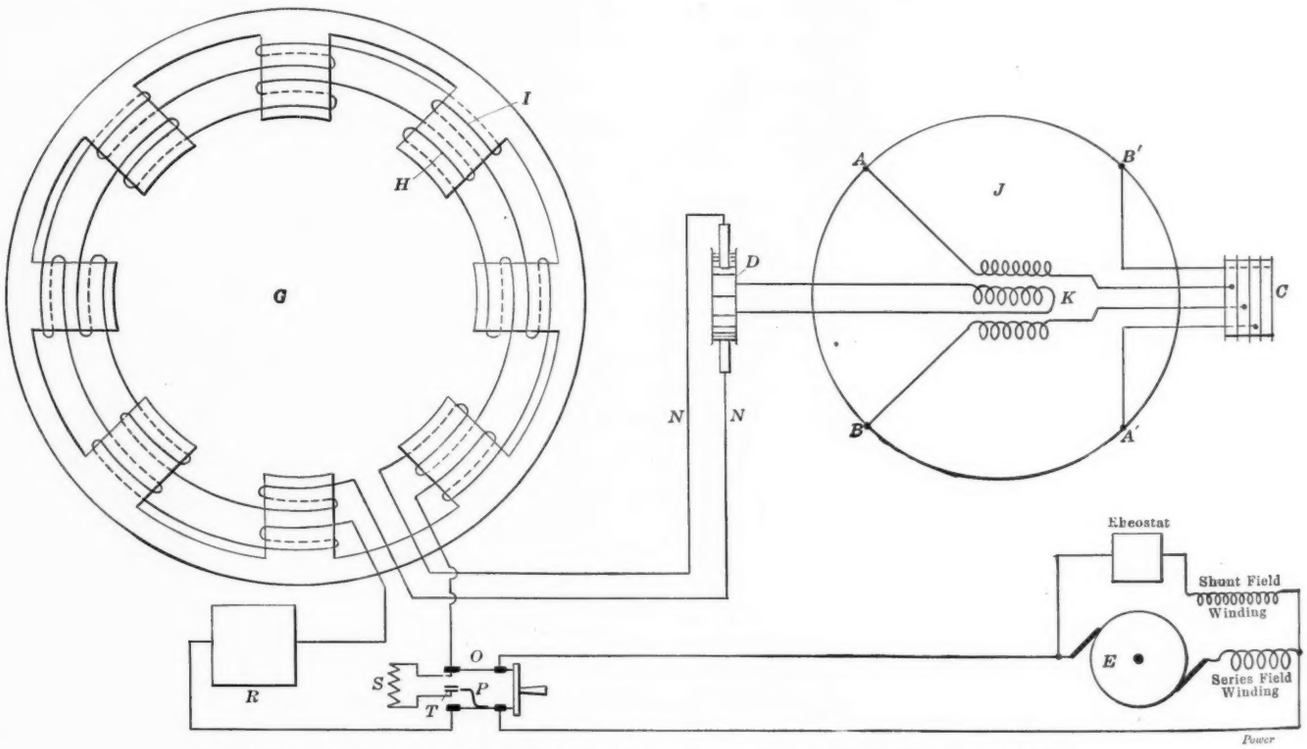


Fig. 1. Diagram of Circuits and Connections of Compounded Alternator

load is reduced the opposite effect is produced.

As the exciters were loaded only to about 60 or 70 per cent. of their capacity, the brushes on the commutator *D* were short-circuited, as shown at *W*, Fig. 2, thereby cutting out the auxiliary series field winding and leaving the field magnets to be magnetized only by the current from the exciters. The machines were run in parallel for several days in this condition, in order to note if any

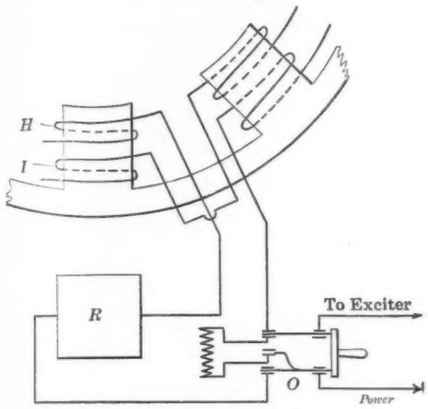


FIG. 2. FIELD WINDINGS COMBINED

cross currents developed. As they operated without any difficulty in this condition the series field winding *H* was disconnected from the commutator *D* and connected in series with the main winding *I*, as shown in Fig. 3. The leads *N N* from the brushes were removed, also the brushes, brush-holders and "jumper" *W*,

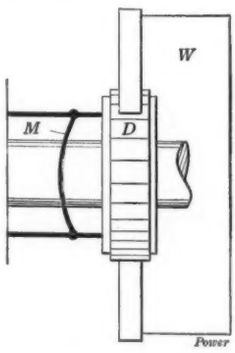


FIG. 3. SHORT-CIRCUITED RECTIFIER

Fig. 3. This made it necessary to short-circuit the segments of the commutator *D*, as shown at *M*, in order to short-circuit the transformer *K* and prevent it from opposing the flow of current in the armature leads *A* and *B*. After this change was made no cross current showed up except such as was caused by poor regulation by the engine governors.

In connecting the two field windings in series it was necessary to make sure that the windings would not oppose each other in their magnetizing effect. Of course, the auxiliary winding could have been left out of circuit entirely, but by connecting it in series with the main winding its magnetizing effect was utilized and helped out the exciters.

Care and Maintenance of Induction Motors

(Concluded)

BY R. H. FENKHAUSEN

Controllers, autostarters, oil switches and circuit-breakers require little attention beyond an occasional adjustment of the fingers and contact clips which can be attended to by the motor inspector on his weekly visits. There is, however, a tendency among machine operators to tinker with the controlling apparatus whenever the motor does not run to suit them, and this tendency if not checked will result in excessive maintenance expense. Fig. 10 shows a simple device by which a controller or starter front may be locked to prevent tampering by unauthorized persons. It consists merely of a strap-iron bracket inside the cover, fitted with a lock staple protruding through a slot in the cover. The adjusting screw on a circuit-breaker may be locked in a somewhat similar manner. Fig. 11 shows the method used on the I-T-E circuit-breaker which may be adapted to other makes by exercising a little ingenuity.

A stock of repair parts for all motors and controllers should be kept on hand, as the time lost in waiting for a new part to be made will mean a loss of many times the cost of the part, and in addition it will be found that small repair parts made in the shop cost from ten to twenty times the price asked by the manufacturer, who can usually, by one stroke of a special die, form a repair part which even a skilled mechanic would require several hours to make by hand.

The following spares are recommended for standard motors: Twelve stator coils of each type used, 2 spare bearing sleeves of each size, fish paper and oiled linen for coils.

Wound-rotor induction motors require in addition: One set of gauze insertion carbon brushes of each size, 1 set of tension springs for brush-holders, 100 pigtails for brushes.

For autostarters and controllers: Ten gallons of transformer oil; 1 set of drum segments, 12 contact fingers, for drum type; 1 set of blades, 1 set of contact clips, for switch type; 1 operating handle for each type.

Spare autotransformer coils need not be carried because there is usually a spare starter available and it can be substituted in less time than would be required to change the coils, especially if connectors are used as recommended in a former article.

Almost all large plants have spare motors for special purposes and to insure against delay from breakdowns, and as these motors receive hard usage and are frequently moved, they are more liable to damage than motors in regular service. The principal source of trouble comes from neglect to empty the oil wells before moving the motor; as the motors are seldom kept level in transit, the oil is spilled all over the windings. Instead of removing the drain plug and drawing the oil off into an old can, where it will gather dust, as is the usual practice, the devices shown in Figs. 12 and 13 may be used and will save time and oil. The drain plugs on all spare motors and portable sets are removed and pet cocks substituted. The male end of a ⅜-inch ground union is screwed on to each pet cock. The syringe shown in Fig. 13 may

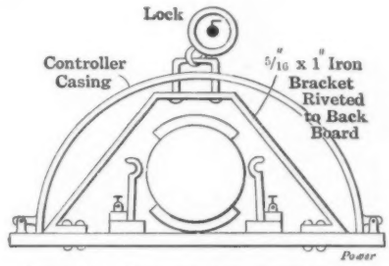


FIG. 10. CONTROLLER-COVER LOCK

be made up from standard fittings and a cup leather fitted to the piston; the female end and union nut of a ⅜-inch ground are attached to the nozzle of the syringe. As many more male ends are required than female, we made up ours

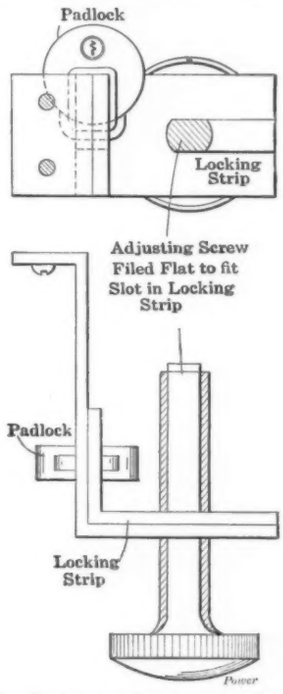


FIG. 11. LOCKING ADJUSTING SCREW ON CIRCUIT-BREAKER

in the shop to avoid breaking up complete unions. Before removing a motor a syringe is attached to the pet cock of each bearing, the cocks opened and the oil drawn out of the wells. When the motor is set at its new location the oil is forced back into the wells by means of the syringes, the cocks closed and the syringes uncoupled. Portable blower and

pumping sets for use aboard ship, where they are handled many times a day and hoisted from deck to deck, have syringes permanently attached to each motor.

In addition to the weekly cleaning, motors should be blown out with an air hose every sixty days. To enable the air to dislodge all greasy dirt a special nozzle similar to Fig. 14 should be used; this increases the force of the blast, and the shape of the orifice gives a thin, flat jet that literally "slices" off the grease and dirt and does particularly effective work in the airgap and between the coils and end connections.

When changing motors from a floor to either wall or ceiling suspension, it is, of course, necessary to shift the heads through 90 degrees or 180 degrees in order to bring the oil wells into the proper position. When the heads are of the pattern shown in Fig. 2, with oil-well covers embracing an arc of 90 degrees from the vertical center line, there is danger of a careless employee replacing the heads with the oil wells on the side instead of on the bottom. On the smaller sizes of motors up to 5 horsepower many manufacturers make the heads perfectly symmetrical and the oil-well cover is the only indication of top and bottom. A little consideration will show that displacement of the head through an angle of 90 degrees will leave the cover in the same relative position, although on the other side of the center line. The only sure way is to look at the oil rings.

FIG. 14. A GOOD AIR NOZZLE

Most motors of 5 horsepower and under have heads bolted on with small hexagon-head cap screws often located in such close quarters that it is difficult to insert or remove them without using a socket wrench. As wrenches of this type are not apt to be at hand when needed in a hurry, a great saving of time and patience will result if the heads of the cap screws be slotted with a hacksaw. The cap screws may then be driven home with great ease by means of a brace and a screwdriver bit.

A Simple Solution of a Complex Resistance Problem

BY J. M. ROW

The accompanying problem in divided circuits is generally considered to be very difficult to solve without algebra, but I have found that it may be worked out by the use of nothing higher than simple arithmetic.

In the diagram *A*—*B*, *A*—*C*, *B*—*C*, *B*—*D* and *C*—*D* represent five wires having a resistance of 4, 11, 3, 7 and 2 ohms, respectively. It is desired to find the joint resistance from *A* to *D*.

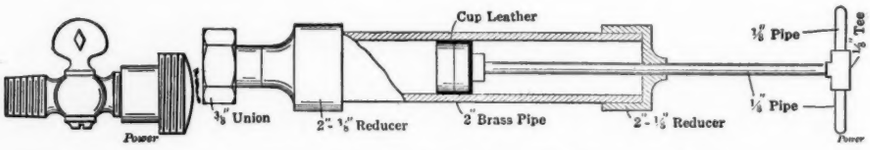


FIG. 12. FIG. 13. PUMP FOR REMOVING OIL

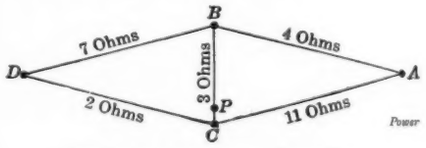
The first thing is to determine whether or not any current will find its way through *B*—*C* when passing from *A* to *D;* and, if so, to determine in which direction it will flow. This may be ascertained by a simple statement of proportion thus: if $\frac{4}{11} = \frac{4+7}{11+2}$ then no current will flow through *B C* and the solution is very simple, being the same as though the wire from *B* to *C* had been removed. But if $\frac{4}{11}$ is greater than $\frac{4+7}{11+2}$ the current will flow from *C* to *B*, while if it is less the current will flow from *B* to *C*. This is all based on the assumption that *A* is connected to the positive terminal of the source of power. It is evident in the case assumed that current will flow in the cross wire from *B* to *C*.

Now assume a potential from *A* to *D* sufficient to cause one ampere to flow from *B* to *C*. It is evident, then, that one ampere comes from *A* to *B*, then goes to *C* and thence to *D*. Of course, more current will flow from *A* to *D* than this one ampere, but the remainder of the current may be neglected for the present. Since one ampere is flowing from *B* to *C* it follows that the potential at *B* is 3 volts higher than at *C*. Now suppose this wire to be cut at some point, as at *P*. This will interrupt the flow of that one ampere from *A* to *B*, to *C* and to *D*. Now that one ampere from *A* to *B* caused a drop of 4 volts which is now available for sending current from *A* past *B* to *D*, and it will produce $\frac{4}{11}$ amperes. It is evident that we have lost one ampere and gained $\frac{4}{11}$ ampere; the net result is a loss of $\frac{7}{11}$ ampere in that branch. This decreases the drop from *A* to *B* by $\frac{4 \times 7}{11} = \frac{28}{11}$ volts. That is to say, the potential at *B* is $\frac{28}{11}$ volts higher than when the circuit is closed from *B* to *C*.

In the lower branch, when the one ampere was interrupted which flowed from *C* to *D*, it decreased the drop by 2 volts, which became available for sending current from *A* through *C* to *D*, and will produce $\frac{2}{13}$ ampere; this, flowing from *A* to *C* will increase the drop in that branch by $\frac{22}{13}$ volts. This leaves the potential of *C* $\frac{22}{13}$ volts lower than when the circuit was closed at *P*. Now *B* was 3 volts above *C* and increased $\frac{28}{11}$ volts, while *C* decreased $\frac{22}{13}$ volts. This makes a potential difference across *B C* of $3 + \frac{28}{11} + \frac{22}{13} = \frac{1035}{143}$ volts when *B C* is cut and the voltage from *A* to *D* is just sufficient to cause one ampere to flow from *B* to *C* if the circuit were closed.

With a potential of one volt from *A* to *D* and the circuit open at *P*, the drop from *A* to *B* is $\frac{4}{11}$ volt, while from *A* to *C* it is $\frac{11}{13}$ volt. The potential difference from *B* to *C*, then, is $\frac{11}{13} - \frac{4}{11} = \frac{79}{143}$ volt when the potential form *A* to *D* is one volt; while with some unknown voltage from *A* to *D* which was sufficient to cause one ampere to flow from *B* to *C* a potential difference of $\frac{1035}{143}$ volts was maintained from *B* to *C*. This unknown potential may be found now by dividing $\frac{1035}{143}$ by $\frac{79}{143}$, and it equals $\frac{1035}{79}$.

With the wire *B*—*C* still open the current through the upper branch of the diamond is $\frac{1035}{79} \div 11 = \frac{1035}{869}$ amperes and that through the lower branch is $\frac{1035}{79} \div 13 = \frac{1035}{1027}$ amperes. It was previously ascertained that opening the cross wire decreased the current from *A* to *B* by $\frac{7}{11}$ ampere and increased the current from *A* to *C* by $\frac{2}{13}$ ampere. Then with the circuit closed from *B* to *C* the current from *A* to *B* is $\frac{1035}{869} + \frac{7}{11} = \frac{1588}{869}$ amperes; while from *A* to *C* it is $\frac{1035}{1027} - \frac{2}{13} = \frac{877}{1027}$ ampere. The total current flowing from *A* to *D* is $\frac{1588}{869} + \frac{877}{1027} = \frac{2,400,389}{892,463}$ amperes, and as this is produced by a voltage of $\frac{1035}{79}$ the joint resistance from *A* to *D* must be $\frac{1035}{79} \div \frac{2,400,389}{892,463} = 4.889$ ohms.



PROBLEM IN JOINT CIRCUITS

Gas Power Department

Modification of the Humphrey Pump*

By Herbert A. Humphrey

This paper deals with the subject of Humphrey pumps from the broad point of view, omitting all details as to types of valves and valve gear except in one instance just to illustrate the kind of device which has been found suitable. Also, to make the paper more complete in itself, some of the old ground is covered again.

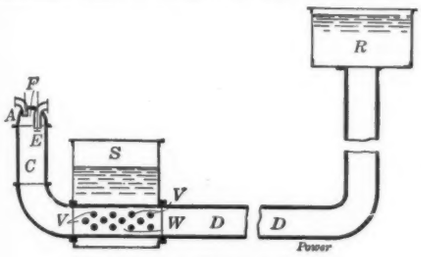


Fig. 1. The Simple Pump

Take the simplest case of Humphrey pump, as illustrated in Fig. 1. Water has to be raised from a supply tank *S* to an elevated tank *R*. The water-valve chamber *W* is equipped with valves *V* opening inward. In the top of the combustion chamber *C* are an inlet valve *A* and an exhaust valve *E*. A simple interlocking gear is arranged between these two valves so that when the valve *A* opens and closes it locks itself shut and releases the valve *E*, and when the valve *E* opens and closes it locks itself shut and releases the valve *A*. Consequently, when suction occurs in the chamber these valves open in turn.

Imagine a charge of gas and air to be compressed in the top of the chamber *C*, and fired by a spark plug which projects through the top casting. All the valves are shut when the explosion occurs, and the increase in pressure drives the water downward in the pump and sets the whole column of water in the discharge pipe *D* in motion. The column of water attains kinetic energy while work is being done upon it by the expanding gases, so that when these gases reach atmospheric pressure the column of water may be moving with considerable velocity. The motion of this column of water cannot be suddenly arrested, hence the pressure in *C* falls below that of the atmosphere, the exhaust valve *E* opens and also the water valves *V*. Water rushes in through the water valves mostly to follow the moving column in the pipe *D*, but partly to rise in the chamber *C* in an effort to reach the same level inside the chamber as exists in the suction tank.

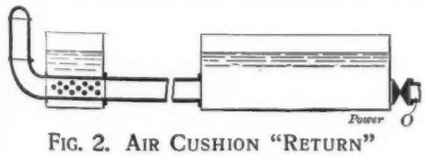


Fig. 2. Air Cushion "Return"

When the kinetic energy of the moving column has expended itself by forcing water into the elevated tank, it comes to rest, and there being nothing to prevent a return flow, the column starts to move back toward the pump and gains velocity until the water reaches the level of the exhaust valve, which it shuts by impact.

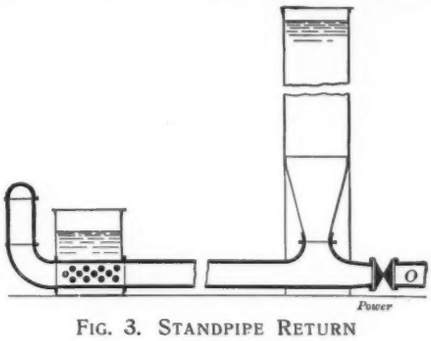


Fig. 3. Standpipe Return

A certain quantity of burnt products is now imprisoned in the cushion space *F*, and the energy of the moving column is expended in compressing this gas cushion to a greater pressure than that due to the static head of the water in the tank *R*. Hence a second outward movement of the column results, and when the water reaches the level of the valve *E* the pressure in the space *F* is again atmospheric, and further movement of the water opens the valve *A* against a light spring and draws in a fresh charge of gas and air. If there were no friction the water would fall to the same level as that from which the last upward motion started, but the amount of combustible charge drawn in is slightly less than this movement would represent. Once more the column of water returns under the elevated tank pressure and compresses the charge of gas and air, which is then ignited to start a fresh cycle of operations.

The action of the pump is not altered if, instead of delivering into an elevated tank, it discharges into an air vessel, as shown in Fig. 2, or into an open-top stand pipe or tower, as shown in Fig. 3, and both of these arrangements are useful if a continuous flow from the outlet *O* is desired. In many cases the pump can be placed directly in a concrete pit, which serves also as a foundation, and the discharge can be by an inclined pipe, as shown in Fig. 4, where the water-valve box is shown vertical.

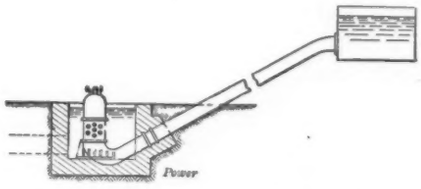


Fig. 4. Vertical Water-valve Box

In the simple form of pump the degree of compression of the combustible charge prior to ignition depends on the hight to which the water is raised, and exceeds the static equivalent of the head. This can be made clear by reference to Fig. 5, where water returning under a pressure *A B*, and moving through a stroke *A D*, does work in compressing the gas equal to *A B C D A*. The gas is compressed from *A* to *E*, and the work of compression is *A E D A*; consequently the shaded area above the line *C B* must be equal to the shaded area below it, and *E*, the maximum compression pressure, is much above *C*. This result is obvious if one remembers that the kinetic energy acquired by the liquid column on its return flow is utilized in compressing the combustible gas while the compression brings the column to rest.

The same considerations enter into the question of the cushion pressure attained, but here we are dealing with the compression of a volume of gaseous fluid which occupies the clearance space only, and the stroke of the water column is greater in proportion, because for the first part of the stroke exhaust products are being expelled and no compression occurs. Without going into the mathematics of the subject, it is sufficient to say that

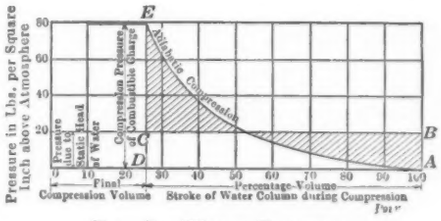


Fig. 5. Work Diagram

*Abstract from a paper read before the Manchester (Eng.) Association of Engineers.

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

the cushion pressure rises rapidly as the hight to which the water is lifted increases, and at the maximum lift of about 40 feet to which the simplest type of pump is limited, it may exceed the explosion pressure when using producer gas, or may approximately equal the explosion pressure when working with town gas or gasolene.

No simple means of indicating the work done by the expanding gases has been found, so that the indicated horsepower of a Humphrey pump cannot be directly ascertained, but indicator diagrams showing pressures on a time basis are interesting. Such an indicator diagram is shown in Fig. 6; this was taken from a 45 water horsepower four-stroke pump, lifting to 39.4-foot head and working on town gas. The figures relating to this diagram are:

Compression pressure at ignition, 56.5 pounds per square inch.
Explosion pressure, 163 pounds per square inch.
Cushion pressure, 185 pounds per square inch.
The pump was delivering 306 gallons per working stroke.

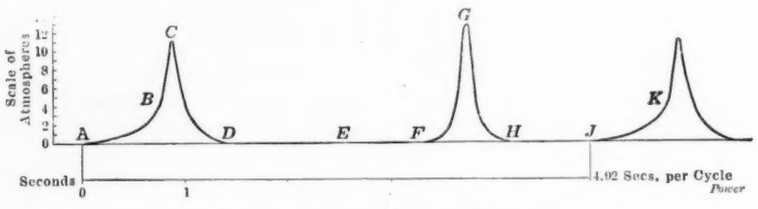


FIG. 6. TIME-PRESSURE DIAGRAM

The diagram may be explained as follows:
Compression of charge from *A* to *B*.
Explosion from *B* to *C*.
Expansion from *C* to *D*.
Taking in water and scavenging air and partly exhausting, *D* to *E*.
Return stroke completing exhaust till valve shuts, *E* to *F*.
Cushioning, *F* to *G*.
Cushion expansion, *G* to *H*.
Intake of new charge, *H* to *J*.
Next compression of charge, *J* to *K*.

The stiff spring used does not permit the suction to be shown on this diagram, but the partial vacuum at the end of cushion expansion may momentarily be as much as four pounds per square inch, and the water suction may at first be equal to two or three feet of water, both dying down to atmospheric pressure during the respective strokes.

The author has dealt at some length with the four-stroke cycle pump, because its construction is simple, and it is the key to all the others. The four unequal strokes which result from the free oscillations of the water column give a cycle which is superior to the Otto cycle, and is in fact the theoretically perfect heat cycle described in textbooks.

Figs. 7 and 8 illustrate the valve gear of the pump which was shown in operation at the Brussels International Exhibition. The bolt *B* sliding horizontally must lock either the admission valve *A* or the exhaust valve *E* by engaging under one of the collars *a* and *e*, which are fixed on the stems of their respective valves. The bolt is urged right or left, according to whether the spring s_1 or s_2 is pulling the hardest, and this again depends on whether the link *l* to which the springs are attached has been shifted to the right or left. Suppose the exhaust valve opened last, then its washer *m* by pressing against the cam arm *p* will have moved the system *p l q* so that it leans to the right, in which position it is retained by the tension of the spring s_2. This puts tension on the spring s_1 and loosens the spring s_2; the bolt *B*, therefore, tries to move to the right, but until the exhaust valve shuts it can only press against the collar *e*. However, when the valve *E* comes back to its seat the bolt instantly locks under *e*, holding the valve *E* shut and releasing the valve *A*, so that the next time suction occurs in the combustion chamber, *A* only can open. Precisely the same kind of action occurs when *A* shuts; it becomes locked and *E* is released again. Thus the valves *A* and *E* are automatically allowed to act alternately, the difference between them being that *E* remains opens till shut by the rising water but *A* shuts under the action of its supporting spring as soon as the suction in the chamber permits the spring to lift the valve to its seat.

The scavenging valve *J*, shown in the plan of the combustion head, Fig. 8, operates at the end of each expansion stroke, its locking and release periods corresponding to those of the exhaust valve; they are made simultaneous by a lever pivoted at *K*, and operated by a pin on the bolt *B*. If the water could rush in fast enough when the pressure falls to atmospheric, there would be no scavenging action, but the incoming water has to be accelerated, and that gives rise to sufficient suction to effect the desired scavenging. In the exhaust outlet there is a light nonreturn valve, to prevent burnt products being drawn back into the chamber.

STARTING THE PUMP

If a pump when working is stopped by switching off the ignition current it will continue its cycle up to the point of compressing the new charge, and then pull up, after a few oscillations of the water column, with the fresh charge ready for ignition, to again start the pump when the current is switched on. In starting up from "all cold" a single charge of air is introduced against the head of water by a hand pump or small compressor, and the exhaust valve is opened by a hand lever to allow the air to escape. This starts the oscillation of the column which automatically shuts the exhaust valve, cushions, draws in a new charge, compresses and fires it, and so the pump gets away under load.

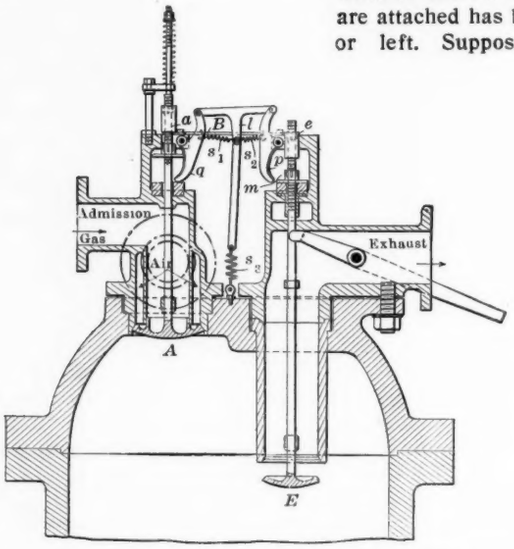


FIG. 7.

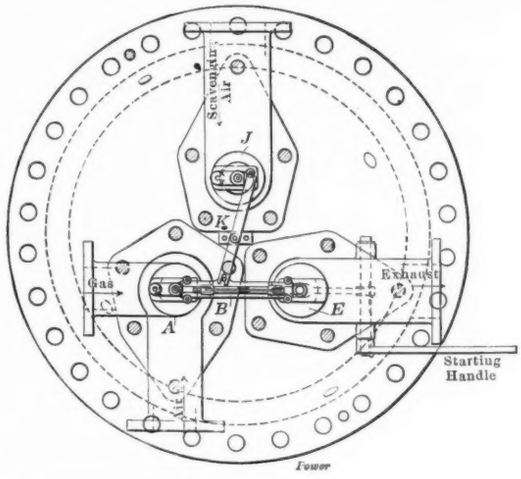


FIG. 8.

VALVE MECHANISM OF THE STANDARD SIMPLE PUMP

High Compression with Low Cushion Pressures

Without much sacrifice of simplicity the cushion space may be kept separate from the combustion chamber, and can then be made relatively large because the products or air in the cushion do not then mix with the new charge. This forms one way of keeping the cushion pressure low. If an internal bell be used to provide a separate cushion of burnt products, a slight modification of the gear is required to control the three valves. Instead of a scavenging valve, there would be a valve *J* opening into the top of the internal bell, as shown in Fig. 9, and on its spindle a loose nonreturn valve *Q*. The valve *J* is released to fall at the same time the admission valve opens, but the valve *Q* prevents any inlet into the top of the bell.

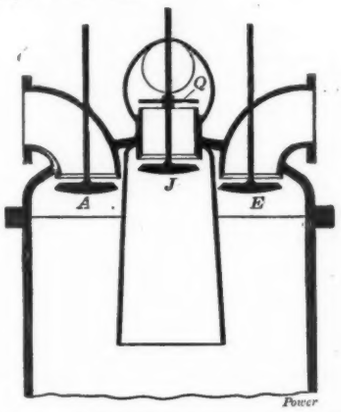


Fig. 9. High-compression Pump Chamber

The action is as follows: When explosion occurs the bell is full of water and all valves are shut. Expansion below the open bottom of the bell allows the water to escape from the bell, which becomes filled with burnt products. On the first instroke of the water the exhaust valve *E* is open as usual and allows the products to escape, but the valve *J* is shut, so that the burnt products in the bell are trapped, and suffer compression by the rising water, which clears away all the other burnt products outside the bell. The compressed cushion in the bell now expands and the second outstroke occurs, during which the suction opens the admission valve *A* to let in the fresh charge as usual, and the valve *J* also opens.

The second outstroke, being shorter than the first, does not lower the water to the bottom of the bell; therefore the pure combustible charge outside the bell does not mix with the products inside. Lastly, the second instroke occurs, when the water rising in the bell forces the exhaust gases out past the nonreturn valve *Q* until the water shuts the valve *J*; the return column of water, having gained velocity depending on the volume of gases discharged from the bell, compresses the pure combustible charge outside the bell. Since the volume of the bell is not limited, the water column may attain a high velocity and so give a high compression pressure, insuring good efficiency.

New High Pressure Pumping Station

By A. D. Blake

The new high-pressure pumping station which has just been completed in connection with Philadelphia's high-pressure fire system, places that city in a position to successfully cope with most any exigency that might arise. The new station is located at Seventh and Lehigh streets and furnishes fire protection to the Kensington and Richmond manufacturing districts. The water supply is taken from the Fair Hill reservoir as the station is some distance from the river.

In many respects the new station is similar to the Delaware avenue station, which has been in successful operation since 1904, but contains many improvements over the earlier installation. Like the Delaware avenue station, gas engines are employed to drive the pumps, the gas supply being taken from the city mains. In choosing gas engines in preference to electric motors for driving the pumps it was claimed that the gas supply in Philadelphia is more dependable than the electric service and, moreover, with the former the city is obliged to pay for only the actual amount of gas that is consumed, whereas, if electricity had been selected, the city would have had to guarantee and pay for a definite minimum amount whether consumed or not.

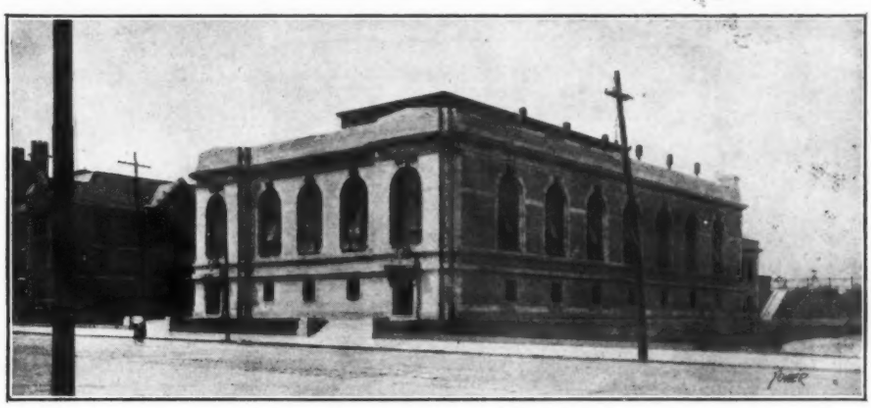


Fig. 1. Exterior View of Station

Fig. 1 shows an exterior view of the station, the appearance of which reflects much credit upon the designers and forms an architectural addition to that section of the city.

The capacity of the plant is 12,000 gallons of water per minute delivered at a working pressure of 300 pounds per square inch to the high-pressure mains which are entirely separate from the city's low-pressure mains. The equipment consists of ten 300-horsepower, 18x22-inch, three-cylinder, four-stroke, Westinghouse, vertical gas engines running at 200 revolutions per minute, each connected through a single 5 to 1 reduction gear to an 11½x12-inch vertical, triplex, double-acting Deane pump. These units are arranged along the main floor in two rows, as shown in Fig. 2. There are no clutches between the engines and pumps, and the former are started by compressed air. The pumps are started under friction load, the water being bypassed to the suction; when the desired speed has been reached, the motor-driven bypass valve is automatically closed, and the pump is ready for service. A relief valve attached to each pump diverts the water back into the suction line when the pressure rises above 300 pounds.

The pumps are of massive construction and when running are unusually free from vibration and noise. The crank shafts are of a single piece of heavy forged steel with the cranks set at 120 degrees. The connecting rods are heavy steel forgings of the marine type with both ends adjustable. The valve chests and cylinders are cast separately so as to localize any damage that might occur and facilitate quick repairs. Both the suction and the discharge valves are located above the highest position of the piston, which insures the pump being primed at all times.

In addition to the ten main units there is a 125-horsepower gas engine of a design similar to the larger ones. This is belt connected to a 20-kilowatt, 110-volt, direct-current Westinghouse generator and to a 9½x5x10-inch Hall tandem air compressor. The generator supplies current for operating the valves on the discharge lines, for the ignition spark in the gas-engine cylinders and for station service. The compressor supplies air at 200 pounds pressure to ten cylindrical tanks, connected in parallel, which furnish compressed air for starting the gas engines. The admission of this air is controlled by a lever-operated valve located at each individual engine. Hence, in order to put a unit into service it is necessary for the operator to go over to that unit.

By an arrangement of cam-operated

valves the compressed air is admitted only to the cylinder whose piston happens to be up.

Precautions have been taken against failure of the compressed-air supply or of the direct current, due to a breakdown of the smaller gas engine, by the installation of a motor-generator set, running off the two-phase city-power circuit. This set supplies direct current in duplicate of the generator previously mentioned, and, in addition, the motor is belted to an air compressor which is an exact duplicate of the compressor belted to the gas engine. In order to carry the extra load of the compressor, the motor is rated at 50 horsepower, although the generator has a capacity of only 20 kilowatts.

To further insure ignition current, each unit is equipped with a ½-horsepower motor-generator set running off the city circuit, and a storage battery, the battery being charged by the small motor-generator set. This makes four sources of supply for ignition current: the generator driven by the gas engine, the large motor-generator set, the small motor-generator sets and the storage batteries, either one of which may be switched into circuit by a small panel-board located at each unit. Each gas-engine cylinder is equipped with duplicate spark plugs, either of which may be used; this obviates a cylinder being put out of service should the spark plug fail while the engine is running.

FIG. 2, INTERIOR OF STATION SHOWING MAIN UNITS

All the piping is located in the basement, which is light and affords plenty of head room. The water supply is brought from the reservoir in two 24-inch pipes, each of which supplies five of the main pumps. These pipes terminate in a U-connection into which is placed a valve; hence, should a break occur in one of these lines, it may be shut off by a valve located at its entrance to the building, and by opening the cross connection, all the pumps may be fed by the other line.

The discharge consists of two 20-inch branches, one running east and the other west from the station. As in the case of the suction lines each of these discharge lines is fed by five of the main pumps and by means of a cross connection and motor-driven valves the discharge from the whole ten pumps may be thrown into either main. Each discharge main is connected to an air chamber which takes up any shocks in the line. A bypass from each supply pipe to the corresponding discharge main, which is fitted with a check and gate valve, insures the high-pressure mains being full of water at all times. This prevents any undue shock which would result if the mains were empty and the water suddenly turned into them at a high pressure.

By means of indicating lamps on the switchboard the operator is able to tell when the motor-driven valve on each branch from a pump to the discharge main is wide open. If the lamp indicates that this valve is wide open and the pressure is not up to the required amount, then it is obvious that another unit must be thrown in.

The jacket water for the cylinders may be taken from the city mains, from the reservoir or from the high-pressure discharge mains. In the latter alternative the water passes through a reducing valve which reduces its pressure from 50 to 70 pounds. Each system of piping in the station is painted with a distinctive color so as to obviate any confusion.

By duplicating many of the more important pieces of apparatus and by providing various sources of supply for the electric current, the chances of the system being put out of service, through failure of any part of the equipment, has been reduced to a minimum. It has been demonstrated that full pressure can be placed upon the system within forty seconds from the time the alarm is received.

Explosion in a Blast Furnace Power Plant

BY OSBORN MONNETT

An explosion of blast-furnace gas occurred about 11 a.m. on Sunday, November 13, at the Youngstown, Ohio, works of the Carnegie Steel Company, demolishing a gas holder said to have had a capacity of 45,000 cubic feet, partly destroying several mill buildings, tearing a hole in the tile roof of the blowing-engine house, and breaking windows in houses and business buildings for blocks around. A number of employees had narrow escapes from death and it is probably owing only to the explosion happening on Sunday that some fatalities were not recorded.

The gas, of approximately 80 B.t.u. per cubic foot, used at this plant for power purposes is taken from furnaces Nos. 5 and 6, each of 600 tons capacity, passed through the usual complement of washers to the engines. The blowing-engine equipment consists of four 3500-horsepower gas-driven units, and there are two gas-driven electric generators each of 2000 kilowatts capacity.

The gas was stored in a holder with a series of deflectors, for the purpose of mixing the gas and insuring a more nearly uniform quality at the engines.

For some time, owing to lack of work, it has been the practice to shut down furnaces Nos. 5 and 6 on Friday of each week, allowing them to stand with fires banked until Sunday, when they were started up again in time for the Sunday night shift. This experiment, it is understood, has been watched with interest by steel men, not all of them being agreed as to the feasibility of the plan.

Standing idle for 48 hours each week gave opportunity for air to percolate into the gas system in various ways, and it has been usual, upon resuming operations, to thoroughly bleed the mains before starting the gas engines, in order to purge the system of any air it might contain. On this occasion, however, it seems that this precaution was either forgotten or neglected and that, consequently, there was still a mixture in the mains capable of propagating a flame. The engines, it is reported, were just being started and only a few power strokes had occurred, when a roar was heard in the gas mains, being instantly transferred to the holder, which was located just outside the blowing-engine house.

The incident emphasizes the need for the greatest care in the operation of such installations and suggests the desirability of developing some form of flame retarder for the intake mains of large gas engines similar to those now being used in the transfer passages of small two-stroke-cycle gasolene engines.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

A Simple Filing Method

To the operating engineer in any plant the problem of taking care of catalogs and valuable printed matter is a big one. He is usually pretty tired when the time comes for him to wash his hands and sit down to a job of indexing, so the indexing is not done. To properly index fragmentary printed matter is a task which even the best equipped man looks at with awe and distaste; hence any system of filing for the man in the engine room must be one that requires no indexing; it must be like the open shelf in the modern library. A few years ago the reader who entered a library had to look through an immense book catalog or a card index in a room apart from the books, access to the books being denied to everyone except the employees. The present practice, however, in the greater number of libraries is to allow the readers to look over the shelves and select their own books. I have found the following method of keeping track of the printed matter, sent so freely by manufacturers, to be very successful.

Cloth- and leather-bound books are set on edge on a shelf in the engine room. The shelf should be kept for the books alone and if men are permitted to get into the habit of using it as a "catch all" for waste, gloves, etc., they soon get in the habit of caring nothing for the books. For printed matter not bound in cloth or heavy bindings, obtain a box 14 inches wide, 10 inches deep and of any length desired. In the bottom of the box bore ¼-inch holes at intervals of 1 inch the entire length on the center line. Out of heavy cardboard, or thin wood, cut a number of sheets 9 inches wide and 13 inches long, with a projection ½ inch high and 2 inches long on top arranged like the guide cards of the ordinary card-index box. Each projection should be past the one in front of it so that for the 14-inch length there will be seven sheets to fill the space. Take a piece of wood or cardboard 9 inches wide by 13 inches long and nail one of the long edges to a piece of board 13 inches long and 4 inches wide. At the end put triangular pieces as braces, so that when the large sheet is upright on the short piece it will stand and be able to resist some pressure. In the bottom piece put three pegs a little less than ¼ inch in diameter an inch apart so that they will fit into the holes in the bottom of the box; this is the "follower."

On the projections of the sheets paste small labels, and mark them, "engines," "boilers," "stokers," etc. Place the follower in place and put the guide sheets in front of it, as nearly in alphabetical order as possible. Place the catalogs in front of the sheet having the label indicating where the catalog belongs. This box will take a 10x12-inch catalog and all intermediate sizes down to 6x9 inches.

When a catalog contains matter on several subjects it can be separated into sections and each section filed where it belongs. When this is done a memorandum should be made on the facing sheets, telling where the other sections are placed. Another way is to have in each compartment a thin book containing a cross-reference index; for example, John Smith & Son issue a catalog dealing with engines, boilers and air compressors. The catalog is filed under "engines" and the word is written in red pencil on the cover so that it will be returned to the proper division if ever taken out of the box. In the divisions reserved respectively for boilers and air compressors will be written in the cross-reference books, "See catalog of John Smith & Son, filed with engines."

The engineer often reads "kinks" and interesting articles in the papers he receives. They are forgotten or cut out and lost because no systematic method of filing is used; for such matter the old-fashioned scrapbook idea is good but this should not be bulky and it is better to have several, one for each subject, than to have one. By removing the alternate leaves the clippings do not cause the books to bulge very much. File these books in the compartments in the same way the catalogs are filed. When the clippings occupy a full page or are partly on one page and partly on another they cannot be pasted but can be filed away in folders just as letters are filed in vertical filing cases.

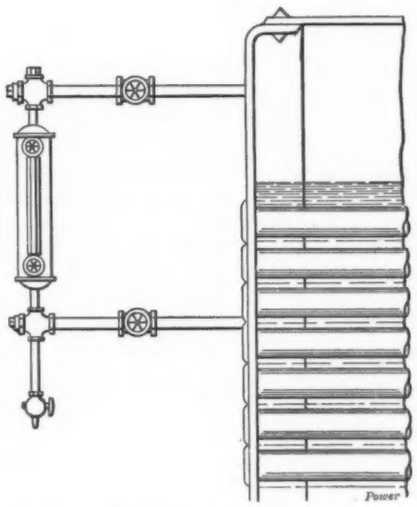
This filing system requires practically no work on the part of the engineer, for the largest part of the work is done when the box and its guides are made. The cover can be lifted and a catalog or clipping put in the box in a very short time and very few require any cross references.

ERNEST MCCULLOUGH.

Chicago, Ill.

Water Gages

There is no appliance used in connection with boilers that needs more attention than the water column. It is true that the safety valve is of great importance in preventing overpressure; but, if a water column, which is depended upon to show a correct and safe level of water in the boiler, fails and misleads, the situation becomes very dangerous. Some columns are not placed upon the boilers so that they will show the correct water level, and this is really more dangerous than a column that is out of repair, as in the former case the water tender is depending upon a certain water level and may be running much below it.



VALVES PLACED IN CONNECTION TO WATER COLUMN

It is not an uncommon thing to find a set of columns placed upon a boiler without cutoff valves between the boiler and the columns. Some engineers claim that no valve should be placed between the boiler and the column, and this theory has some merit, but it seems to be an unsettled question. The only point that can be urged against it is the fact that the valve could drop upon the seat and close off communication with the steam or water space, but this can happen only when the valve stem is in a vertical position, not when in a horizontal position. There is little chance of the valve becoming detached from the stem and closing off the steam or water passage, but one of the main troubles I have found in connection with the use of globe valves is the closing off of the valve by the formation of scale. To avoid this trouble, however, a small gate valve may be used in a horizontal position. There is one great advantage in having the column so that it may be cut off from the boiler, and

that is to make repairs or clean the column while the boiler is hot.

The accompanying sketch shows a method of attaching water columns that has been found satisfactory. By placing the two crosses as shown and by using the gate valves the two plugs *C* and *D* can be removed and a rod run through the pipes to remove any obstructions. A blowoff pipe should always be inserted so that a circulation may be started at any time in the column. Care should be taken in setting the column body with reference to the water line of the boiler. The water line on horizontal return-tubular boilers should be at least three inches above the upper tubes and the gage line should be carefully set to this. Columns are frequently set where this is not carefully observed, in which case the column does not truly indicate the water level in the boiler and this is worse than not having a column.

C. R. McGAHEY.

Lynchburg, Va.

Accepting Advice

Some engineers seem to regard the supply salesmen as a sort of unnecessary evil and almost invariably treat them as such when they are given a friendly call by these "knights of the grip." Although I find it necessary to "get busy" sometimes, I have never yet seen the occasion or the salesman with whom I could not spare the time to sit down and "talk it over," even though quite sure that I needed nothing in his line. Invariably he has something to say which will be of interest and I have often received pointers which proved quite valuable in my work.

There are engineers who seem to be possessed with the idea that when anyone volunteers information that they are considered as a dummy or else the other man just wants to show how smart he is. It always seemed to me that such men held a spite for themselves. I do not mean that a man should believe everything he is told or reads, but if he has any powers of reasoning he can decide for himself the truth or falsity of the assertions. A person should not expect to put into practice only original ideas, for if such were the case, they would have to live to a great age in order to accomplish anything worth while.

I have been a reader of POWER for a number of years and almost invariably upon picking up a new issue turn first to the advertisements to see if there is anything new in the market.

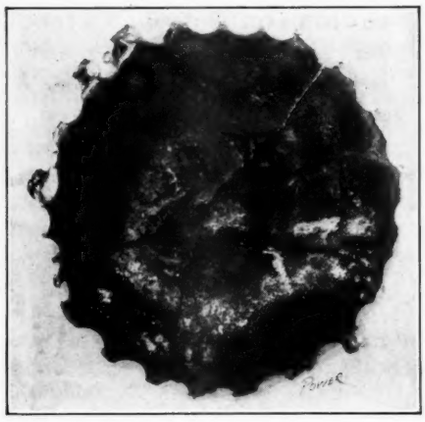
I do not think that engineers are any more obstinate about being shown the errors of their ways than men of other professions, but engineers notice it more because they come in contact with men of their own profession more than with men of other callings.

A. RANSOM.

Wheeling, W. Va.

Laminated Boiler Plate

The accompanying view shows a piece of blistered plate that was cut from the front sheet of a 6x18-foot return-tubular boiler. The plate was ½ inch thick, originally, but after the laminations set in



HOLE BURNED IN LAMINATION

it was split into practically two plates, one ⅛ inch thick and the other ⅜ inch; the thicker being the one through which the hole was burned. The boiler had been in service two years and, although designed for 150 pounds pressure, it was being operated at only 100 pounds.

A. F. ROGERS.

Negaunee, Mich.

Beading Boiler Flue

I would like to have the opinion of POWER readers as to which is the better way to bead flues in a return-tubular boiler. Will they hold better and last longer if they are beaded down flat and tight against the head or should the small space be left which is the natural consequence of light beading when the outside edge of the bead is tight against the head?

L. EARLE BROWN.

Roundup, Mont.

A Solid Gasket

A piston pump which was used to raise water to a tank situated a few hundred feet up the side of a hill was left to the care of a roustabout in the mill, and occasionally the superintendent would stroll down to look it over. One day some imaginary trouble caused them to remove the head of the steam cylinder and when the head was replaced the trouble commenced. Steam was admitted to the cylinder and it made one stroke. The pump was taken apart but to no avail. Finally the roustabout discovered that by throwing over the reverse lever the pump would make two strokes, so in this way the tank was kept full for two or three days.

The superintendent did not care to expose his ignorance but finally decided it would be better to consult a pump expert than to run the pump by hand. When the latter arrived he requested that the cylinder head be removed. "Now, then," he said, "take your knife and make a hole in the gasket at that indented spot." When these orders had been executed the cylinder head was replaced, steam was admitted to the cylinder and the pump ran without a hitch, much to the astonishment of the two bystanders.

When the new gasket had been placed on the cylinder, holes had not been made to allow for the escape of the steam from that end of the steam chest.

EARL E. WEBSTER.

Maxton, Ariz.

Liquid Discharging Device

The accompanying illustrations show a device for ejecting oil or other liquids from a barrel, which I have recently had patented. A metal plug *B* which is threaded and tapered to fit any sized hole is screwed into the bunghole of the barrel

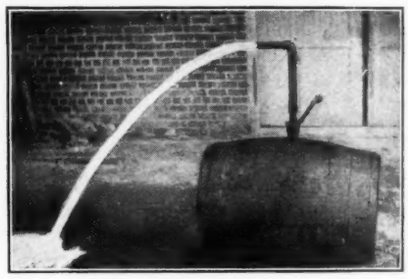


FIG. 1. DEVICE IN OPERATION

to be emptied. Through this plug a delivery tube *D* extends to within a short distance of the bottom of the barrel, and an inlet pipe *F* for compressed air or steam projects into the annular space be-

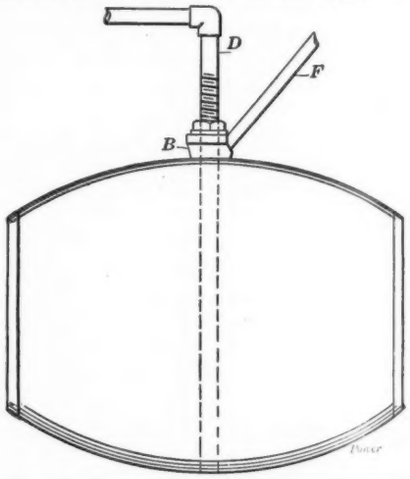


FIG. 2. PIPE PROJECTING INTO BARREL

tween the plug and the delivery tube. When the apparatus has been adjusted, pressure is applied to the surface of the liquid in the barrel through pipe *F* and the liquid is ejected through the outlet pipe.

EARL PACETT.

Coffeyville, Kan.

Size of Indicator Diagrams

I have noted for a number of years that indicator diagrams have been taken almost entirely with the standard length of approximately 4 inches, regardless of the hight of the diagram, and in some cases even longer diagrams have been taken. It is my opinion that the length should have some definite relation to the hight. Diagrams are taken for the purpose of setting the valves or determining the power which is generated by the engine. If they are taken for the former purpose, it is essential for best results that the important events of the stroke be sharply indicated. A long, narrow diagram will result in all of the important events being long drawn out, and make it difficult to determine at what point they occur. The same is true with a short high diagram. On the other hand, one which is nearly square will show up the various events of the stroke to a marked extent, and make it possible to determine accurately where each event takes place. Those taken for the purpose of determining the power require only such a length as will allow of accurate measurement and a sufficient area to be determined within the limits of accuracy of the length measurement. Any greater length than this is valueless. If a length can be determined within 1 per cent., it is long enough, and the area can be determined readily within this length if the diagram exceeds one inch in area. Also, for high-speed operation, the small diagram reduces to a very considerable extent the possible error due to pencil friction, inertia of the drum and other errors incident to indicator work.

HENRY D. JACKSON.

Boston, Mass.

Duplex Pump Trouble

A somewhat puzzling failure of a new duplex pump to work satisfactorily was due to the piston-rod nut in the steam cylinder becoming loose, thereby permitting steam to pass under the packing ring, between the piston head and the follower.

This pump had been in service only a few days when it was noted that one side began to lag in its stroke toward the steam-cylinder end, acting at first as though the discharge valve had become jammed, or the steam valve had become deranged. After examining these parts and finding them in good condition, the steam-cylinder head was removed, and on applying a wrench to the piston-rod nuts one of them was found to have loosened less than half a turn. This was tightened, the head was replaced, and the pump worked in a satisfactory manner.

This experience goes to show that packing rings should be a good working fit in pistons; otherwise an excessive waste of steam will occur without anyone suspecting such to be the case.

The ordinary duplex pump is especially inefficient in point of economy, but when the foregoing conditions occur, or where the pumps are permitted to run with a short stroke, their efficiency is still further decreased.

W. E. RECTOR.

Brooklyn, N. Y.

Remodeling a Pump

Several years ago on taking charge of a plant, I found a bilge or sewerage pump which was supplying a piping system

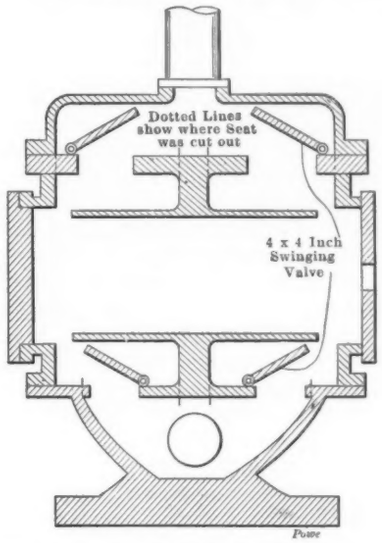


FIG. 1. PUMP BEFORE REMODELING

carrying a pressure of about 40 pounds per square inch. The closing of the valves under this pressure was so sudden that it caused a hammer or pound which made all the piping shake, and several joints leak. I overcame this trouble by remodeling the pump as shown in the accompanying sketches.

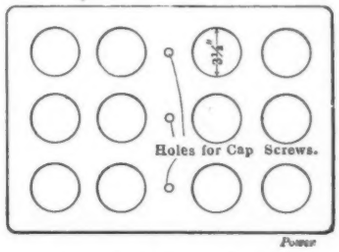


FIG. 2. VALVE PLATE

Fig. 1 shows how the inside of the pump was cut out to afford a less obstructed passage to the valve plates. I had two ½-inch plates, see Fig. 2, cast at a local foundry, and then planed and bored for twelve 3½-inch valves; also, twenty-four 3½-inch valve seats cast of brass and twenty-four studs for them.

The seats were finished in our own shop, being made a driving fit in the plates, and the studs were finished to ⅝ inch. Putting in the plates made it necessary to use two gaskets on each plate, also to procure longer cap screws for bolting the pump together again.

Fig. 3 shows the pump completed with the plates and valves in place. We had to raise the steam end of the pump off of

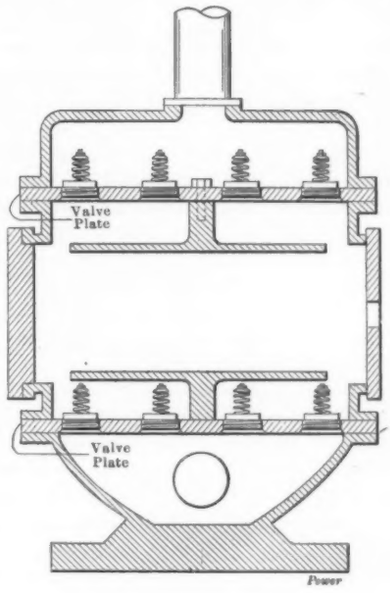


FIG. 3. REMODELED PUMP

the foundation a distance equal to the thickness of the valve plate in order to aline it with the water end. The total cost of remodeling this pump did not exceed $30. And the pump has since been giving entire satisfaction.

W. H. BEISLEY.

Natchez, Miss.

Shortage in Coal Weights

Some of us are very careful to weigh our coal before it goes into the furnace, but how many consider whether the cargo held out as per bill of lading? I have had my eyes opened during the past few months. For instance, I received a new cargo after all the old supply was cleared up, so I did not have to guess how much coal there was already on hand. This cargo weighed up correct in short tons, but it should have been in long tons.

On receiving another cargo of coal, it was dumped into cars of the same dimensions and loaded about the same; one load was carefully weighed, taking the total number of car loads and averaging the weight, the cargo came out 55 tons short, weighing in short tons. On the first cargo, which was 100 tons short, I got no redress as the coal company made an affidavit that its weight was correct.

I believe it would be well to put in a derrick scale and weigh each bucket as it comes from the vessel and then, if the weights are not as per bill of lading, apply to the Interstate Commerce Commission.

THOMAS HAWKIN.

Rockland, Me.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

The Economy of Vigilance in Small Matters

There have been several good hints on the first page of POWER during the last few weeks. The one in the issue of October 25 strikes me as being especially timely. I am inclined to think that when an isolated plant is shut down and service bought from the central station, it is, in a majority of cases, the fault of the engineer and not of the owner. Too often the isolated plant is not operated nearly as economically as it might be if the engineer did his best.

I was once called to take charge of a plant in which the operating expenses seemed to be running too high to suit the management. The man who hired me had a talk with me in regard to the coal bills and other expenses, but, as the plant was new and had only been running three years, he did not seem to know where the trouble lay.

The plant consisted of three fire-tube boilers, 72 inches by 16 feet in size, two high-speed engines and such other equipment as is usually found in a plant of this kind.

After getting fairly "squared away" I began to look for the cause of the high

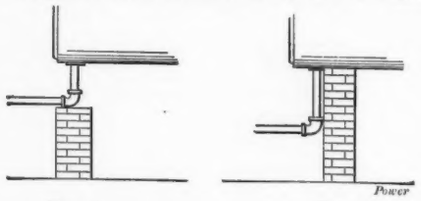


FIG. 1. FIG. 2.

ORIGINAL AND IMPROVED ARRANGEMENT OF BLOWOFF

operating expense. I learned that the man who had formerly had charge of the plant had been accustomed to having the flues cleaned only once a week. I had them cleaned every day, and noticed quite a change in the coal bill. I also learned that they had been in the habit of carrying the water level in the boiler at a hight in the gage glasses of 3 or 10 inches. The bottoms of the glasses were about 4 inches above the top tubes, so that they had been carrying about one foot of water over the top of the flues, thus leaving a very small steam space and getting very wet steam.

The coal pile was some 25 or 30 feet from the boiler and it was the habit of the fireman, when stoking in a fire, to carry each shovelful to the boiler and leave the furnace door open while doing so. I noticed that there was a constant cloud of vapor rising from the hotwell, and on investigation found that the blowoff valves on the boilers were leaking.

The main exhaust pipe from the engines ran under the engine-room floor, out into the boiler room and then up overhead to the heating system. Under the boiler-room floor, I found a ¾-inch valve in the exhaust pipe, which had been put in to bleed the water out of the exhaust pipe. A long valve stem had been attached to this valve and run up through the concrete floor, for the purpose of opening and closing the valve. This valve stem had been broken and the valve left open to the sewer, thus creating a considerable waste when back pressure was used for heating.

The condensation from the heating system came back into a return tank in the boiler room, was pumped from the tank into a feed-water heater with one pump, and from the heater into the boilers with another. There were no automatic float-operated valves on the pumps, so that considerable hot water was lost by overflowing from the heater, it being difficult to regulate both pumps so that this would not occur. On taking the temperature of the returns, I found that they were coming back at a temperature of about 200 degrees Fahrenheit. I therefore arranged to pump the returns directly back to the boilers, using the heater only when taking fresh water.

On opening the door in the back end of one of the boiler settings, the ashes were found to be wet. The blowoff pipe had a brick pier built up under it, as shown in Fig. 1. I found that this pipe had been leaking badly and was burned clear through, nearly the whole way around, only about 1½ inches of metal remaining. One of the other blowoff pipes was found to be in much the same condition. Probably the only reason that the pipe stayed on the boiler was the fact that it was held there by the brick pier. These blowoff pipes were torn out, as were also the brick piers, and extra-heavy pipe put in as shown in Fig. 2.

One of the engines was of the inclosed, self-oiling type. The oil drips, which came out around the piston-rod packing of this engine, were led by a drain to the sewer. This was disconnected and the drips run through a filter and used again. Both engines were indicated and the valves properly set.

While any of the above mentioned items would not create much of a loss by itself, yet, when all were corrected, it resulted in a saving of 25 per cent. on the first year's operation.

It is the little things that count.

R. L. RAYBURN.

Kansas City, Mo.

Water Hammer

In the discussions in POWER on boiler explosions and water hammer, the general opinion is that water hammer due to cutting in a boiler is often the cause of an explosion.

From experiences I have had I came to the conclusion that water hammer does not cause a great many of the explosions credited to it, except in those cases where water has a chance to trap over the stop valves.

I am operating a battery of vertical water-tube boilers. After cleaning and filling, the usual method of allowing the air to escape through the water column while raising steam did not seem to work in this case. As soon as the valve was cracked to cut in the boiler, the snapping, hammering and vibrating would start. To be up 30 feet from the floor on a boiler that is trembling under your feet, and expect something to let go with each snap, is not very pleasant. Extra precautions were taken in raising steam and cutting in the boilers, still we had the same trouble every time a boiler was cut in.

The stop valves were of the angle pattern fitted to the steam nozzle on the boilers. The steam header is 6 feet below the stop valves and connected to them with long-sweep bends. Therefore, there was a perfect drain and I am positive that water was not the cause of the hammering and vibrations. We tapped the steam nozzles and put in ⅜-inch vents to allow the air to escape while raising steam, and that ended the trouble, not only in the boiler room but in the engine room as well, where the vacuum was frequently lost.

This case may have been an exceptionally bad one, but how many plants are there that have perfect vent arrangements for removing the air from the boilers while raising steam? In a great many, the stop valves are placed several feet from the boilers so as to avoid a water trap above the valve. With this arrangement a perfect air trap is formed below the valve and on cracking the valve vibration starts.

My experience has convinced me that

if every engineer would put a vent below and a drain above the stop valves and instruct his men to try both before cutting in a boiler, there would be less explosions credited to water hammer and unequal pressure.

H. J. MISTELE.

Milwaukee, Wis.

Isolated Plant Costs

I have been much interested in the discussion which has been going forward in the columns of POWER recently on the subject of "The Isolated Plant versus the Central Station." I therefore offer my experience in this regard. The subtended figures are authentic and were furnished by the audit department of a large building in the operation of the plant in which I am engaged.

Cost of generating plant in excess of that required for central-station service:

Switchboard and wiring	$1600.00	
1—100 kw. generating unit, including foundation	3575.00	
1—75 kw. generating unit, including foundation	2775.00	
Exhaust pipe trenches and covers	125.00	
Piping for engines, heater, pipe covering, etc	1700.00	
1—200 H.P. boiler and necessary connections	3300.00	
Total		$13,075.00

Expense of operating plant:

12% interest and depreciation against plant cost of $13 075.00	$1569.00
Coal	4987.50
Water	250.00
Wages of chief engineer	1800.00
Wages of ass't. engineer	1144.00
Wages of fireman	832.00
Oil, waste, supplies, repairs, lamp renewals, ashes	975.00
Rent for 2000 sq. ft. of space taken up by generating plant at 30 cents per foot	600.00
	$12,157.50

Electricity, if purchased from the central station at Mr. Tweedy's price of 5 cents per kilowatt-hour, would cost:

356, 500 K.W. hours at 5 cents	$17,825.50	
Building plant	12,157.50	
Saving Over Central Station		$ 5668.50
To this must be added the cost of coal for heating and keeping steam the year around for manufacturing purposes which would be	$ 1900.00	
Labor to take care of plant	1600.00	
Engineers' supplies, water, ashes, repairs, etc	500.00	
Total saving over the cost of central station service		$ 8668.50

CHARLES W. PEACOCK.

Chicago, Ill.

I have been very much interested in the articles appearing from time to time in POWER, in regard to the relative cost of electricity as furnished by the central station and the isolated plant.

In the plant of which I have charge there is 400 horsepower in boilers, a greater part of which is used exclusively for heating purposes. One boiler of 80 horsepower rated capacity, carrying 60 pounds pressure, supplies steam for the laundry, cooking, etc.; the balance of the boilers are operated at from one to ten pounds pressure according to the weather conditions.

Electric current for lighting is furnished by the central station; the average monthly consumption is about 2700 kilowatt-hours at a cost of about 6.5 cents per kilowatt-hour, making a yearly cost of about $2115.

I have been contending for some time that our light was costing too much, but have not as yet been able to convince the management that we could produce the same current much cheaper in a plant of our own.

The heating boilers are operated from seven and a half to eight months each year while the 80-horsepower boiler is run the entire year.

As the plant has to be operated 24 hours per day, the only extra help required would be one assistant engineer. The extra coal consumed would be very little, especially in cold weather, as the exhaust steam would be used in the heating system; even in warm weather, the exhaust could be turned into the water heater which supplies the institution with hot water and in which live steam is now used.

I should like to see some statistics from plants of about the size of this one where current is generated.

H. A. MUNYON.

Syracuse, N. Y.

Installing Globe Valves

I have been interested in the different views expressed in recent numbers of POWER relative to the installation of globe valves.

Ordinarily, I believe that the better way is to put the valve in so that the flow of steam or water, as the case may be, will be against the under side of the disk.

In an instance I have in mind, however, the opposite arrangement would have been preferable. An 8-inch throttle valve of the outside-screw type having a brass spindle was connected to a Corliss engine. Owing to the failure of the trap attached to the drip pipe, this valve was often subjected to the effects of water hammer.

The engineer thought that the engine was not getting enough steam, although the throttle seemed to be wide open. When I entered the engine room I heard the throttle valve clicking away every time the engine took steam. This was conclusive proof that the valve was not fully open, for, had it been, the valve spindle would have drawn the disk tight against the cap and thereby prevented the clicking sound.

At the first opportunity we took the valve apart and found that the spindle was so enlarged, due to the bumping under the disk, that we could not open the valve more than ¾ inch, while it should have been opened 2 inches in order to utilize the full capacity of the pipe.

I put the spindle in a lathe and reduced the thread to work as designed and at the same time took dimensions and later on had a spindle made of mild steel which did not upset.

One morning before starting time the engineer informed me that the engine had "turned over during the night and as hot as if she had been running."

The trouble was that the valve had broken, probably from the effect of water hammer. Not having a spare part on hand, a shutdown of half a day was caused until we made a temporary valve by utilizing a blank flange which was faced off and riveted to a yoke to engage the spindle.

Another example of a similar nature was that of an underwriters' fire pump which had a throttle valve of the outside-screw type wherein the pressure was under the disk. The valve was so located that under certain conditions water hammer would occur. The valve got so that it would not open to more than one-half of its capacity because the spindle was upset from the effect of water hammer as in the first case.

I repaired this spindle by the same process as that applied to the engine valve.

In both instances, I suggested turning the valves, "end for end," and thereby bringing the pressure on the top of the disk which would prevent undue pressure against the spindle in the event of water hammer.

Relative to the throttle valve on the fire pump, in the case of fire, there would have been a chance for controversy if it could be proved that the pump could not get sufficient steam to actuate it to its full capacity. Experience teaches me that inspectors should note the condition of the throttle valves on fire pumps.

J. W. PARKER.

Clinton, Mass.

Metallic Packing

In the power house where I am employed there are three 500-horsepower double-acting gas engines and one 14x16-inch automatic high-speed steam engine, all fitted with metallic packing. We have found that the packing cuts the rods to a taper which necessitates having them turned in the lathe in order to make the packing hold. The gas-engine rods run about one year but that of the steam engine, which was but recently installed, has run only about eight months and will have to be replaced by a new rod, it is cut so badly.

Why do the rods cut to a taper?

The gas-engine packing is lubricated by pumping a good grade of cylinder oil into the caging of the packing by an oil pump which delivers the oil into the pack-

ing on the inward stroke of the piston. Can any reader explain why the rods cut tapering? The tension of the garter springs should be equal at both ends of the stroke.

W. D. MARQUEST.

Alexandria, Ind.

One or Two Boilers

I have been waiting in the hope that some of the contributors to the "One or Two Boilers" question that has had considerable discussion of late, would reply to R. McLaren's statement published in the October 18 number.

I regret that Mr. McLaren did not give some data that would enable us to learn how a saving of over 25 per cent. was effected, for it is quite evident that a man who can cut his fuel expense down 25 per cent. without any outlay for fuel-saving devices on the part of his employer, is wasting time in remaining in a two-boiler plant. If Mr. McLaren had told us what type of boiler he has in use, the boilermaker's rating of it, the grate area and kind of draft employed, also whether conditions were the same in each case, as well as the amount of evaporation under each set of conditions, he might have thrown much light on a question that is of vital importance to every person who has to pay fuel bills as well as to engineers who design or operate the plants.

A question that has claimed the attention of engineers for many years is the advisability of driving steam boilers to their limit. This is a very broad question and any practical experience that will help to its satisfactory solution would doubtlessly be well received.

In many large power plants the boilers are forced to the limit; in fact, this is the almost universal practice in electric-light and power plants. As the engineers and managers of such plants are generally men of intelligence and experience, the natural inference is that the practice is commendable. But in what way is it commendable?

We know that most large power systems are owned by corporations in which the members of the board of directors are elected by the stockholders from time to time. It is the desire of each board of directors to make a good showing regardless of the wear and tear on the plant. A case that came to my notice recently may serve as an illustration. The chief engineer of a large corporation was shown something new in the fuel- and labor-saving line and it appealed to his judgment so that he wanted to try it in one of the plants where fuel was the equivalent of over $8 per ton. He entered a requisition for the necessary funds. He was called on at the first following board meeting and asked if it was necessary to have the device in order to keep the plant in operation. He replied that it was not but that it was evident that the device would pay for itself in reduced operating expenses within a year, and that he thought any investment that would pay at least 100 per cent. *per annum* and probably last as long as the boilers was worthy of consideration, etc. His request was turned down with a statement to the effect that the amount he proposed to spend for betterment could be used to better advantage as dividends. And there you are! It is a question of "frenzied finance" against the judgment of a competent engineer, therefore it does not seem safe to bank too heavily on the practice employed in large power plants, for the question of cost of maintenance is seldom given much attention.

W. H. ODELL.

Yonkers, N. Y.

In the October 18 issue, there is a note by R. McLaren, entitled "One or Two Boilers," in which he does not venture an opinion, but gives the readers his own choice.

Mr. McLaren's experience would lead to two conclusions: First, there should be two boilers installed, for with only one boiler he would have had to shut down his plant when it needed repairs; second, if he can save so much fuel by running one boiler instead of two, he should always run only one boiler.

W. H. KELLER.

Ansted, W. Va.

Thermometers versus Gages

Mr. Peabody's article under the above title in the October 25 issue is very interesting and profitable, but there is room for considerable argument before the steam gage is scrapped and exclusive use made of thermometers.

From the steam tables it is seen that for each equal increment of temperature above 32 degrees Fahrenheit the increment of pressure is a very small fraction of the temperature increment. At 150 degrees Fahrenheit the pressure increment begins to increase more rapidly until at 300 degrees it is equal to the temperature increment. Above this temperature they converge until at 400 degrees a change of one degree is accompanied by a change of 2.8 pounds in the pressure. From this it is clearly evident that with equal divisions on a thermometer and pressure gage, the thermometer would indicate with much greater accuracy the condition of the steam up to 300 degrees. Above 300 the pressure gage would indicate more accurately the condition of the steam.

Since, in actual practice thermometers and steam gages do not have equal divisions, it will be interesting to see what results would be given from the use of an actual steam gage and thermometer. If we assume a 12-inch steam gage and a 10-inch thermometer graduated to pounds and degrees respectively, we shall have about a 20-inch scale on the gage divided into 250 divisions, and a 10-inch scale on the thermometer divided into 370 divisions. This means that the length of a pound division on the gage will be very nearly three degrees on the thermometer. With this arrangement it is evident that the thermometer will be more accurate up to the temperature where one degree change in temperature is accompanied by a change of pressure of one-third of a pound. This is at 220 degrees. Above this temperature the gage would indicate the condition of the steam much more accurately. At 150 pounds, which is about the average of present-day boiler practice, a change of one pound on the gage would be attended by only one-sixth the same linear movement on the thermometer. From the foregoing it is safe to state that a thermometer will give more accurate results up to about atmospheric pressure, but that for boiler work it would hardly seem that a "little glass thermometer" is as convenient to use or as accurate in its results as a "big nickeled gage."

WILLIAM NOTTBERG.

Kansas City, Mo.

Diagrams for Criticism

Referring to the diagrams for criticism in the November 1 issue, Mr. Fryant states that he is reasonably sure that the jagged expansion line was not caused by the indicator. My experience has been that such a line is caused by the indicator only. The peak on the expansion line indicates that the indicator piston was sticking, and the wavy condition of the expansion line indicates the inertia of the instrument.

T. J. HAMMERSLEY.

Milwaukee, Wis.

Quarter Turn Drive

In reply to Mr. Bitterlich's inquiries in the November 1 issue, the following is offered in the hope that it may be of help to him:

All drive-belt formulas are empirical, giving the millwright but a general idea of what is required, and on the subject of quarter-turn drives, in particular, even the latest textbooks are vague. For instance, it is stated that "the line representing the plumb line will be the center of the vertical belt," when, as a matter of fact, in a quarter-turn drive employing only two pulleys, neither belt is vertical.

In order to run a quarter-turn drive successfully, the following points must be observed:

The distance between the near faces of pulleys must not be less than four times the width of belt, and twice that distance is very desirable.

The pulleys must be so placed that a

plumb line, as *E F* in the accompanying sketch, tangent to, touching and dividing the face of one pulley is also tangent to, touching and dividing the face of the other pulley. The belt must lead off each of such faces. In the sketch, the line *E F* passes tangent to the faces *A* and *B*. The belt leads off of the driver at *A* and on to the driven at *C;* leads off the driven at *B* and on to the driver at *D*. Consequently, neither belt is vertical.

Both pulleys should be as small as is consistent with the amount of power to be transmitted.

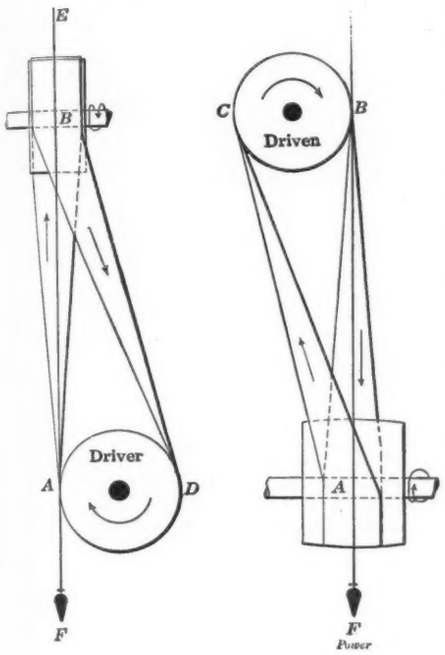


DIAGRAM OF QUARTER-TURN DRIVE

The face of the driving pulley should be crowned at least 30 per cent. more than the usual crowning given to a crowned pulley.

The ratio between the width of the driver and that of the driven should be as 5 is to 3.

About 25 per cent. of the contact is lost with a quarter-turn drive even when the pulleys are of the same size, owing to the imperfect elasticity of the belt. A narrow heavy belt is preferable to a wider and lighter belt, as the latter stretches rapidly on one edge and has a short life.

Leather-link belts give the greatest satisfaction. Being hinged down the middle, such a belt will accommodate itself to the high crowning of the driver, which increases the efficiency of the drive.

A double belt with a narrow upper layer on top and to one side, depending on whether the belt is right- or left-hand running, will also give good results, as the inequality of thickness forces this side against the pulley and makes a good contact.

A heavy-ply narrow rubber belt will give good service if laced smoothly and evenly, with a half turn in the belt, so as to bring each side of the belt alternately against the pulley faces, thus stretching each edge of the belt equally. With a short drive the belt is difficult to throw on, when laced thus, but when run at belt speeds of 1000 feet per minute and more, it will give great satisfaction, both in efficiency and durability.

R. REDDIE.

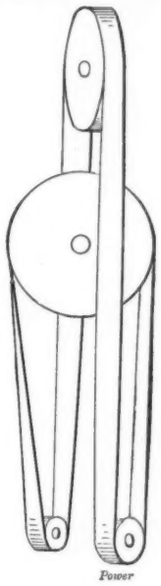
New Market, Tenn.

In the issue of November 1, W. J. Bitterlich asks for information relative to a quarter-turn belt installation.

The minimum distance between shaft centers considered to be good practice is about seven times the width of the belt. The single belt is best used as a basis for calculation. Thus, in Mr. Bitterlich's problem the distance between centers should be kept at least seven times the 6-inch width, even though a change to a 4-inch double belt were made. Designers are not inclined to use this extremely short distance extensively, however, and the following formula is the basis for a permanent and economical operating design:

$$L = C + 15\ W,$$

where L equals the distance between shaft centers in inches; C equals the circumference of the larger pulley in inches, and W equals the width of the belt in inches.



ARRANGEMENT TO SECURE EFFECTIVE DISTANCE BETWEEN CENTERS

With this formula the distance between centers in Mr. Bitterlich's problem should be about 12 feet. The effective shaft distance for vertical quarter-turn drives may be obtained by using idlers on the floor, as indicated in the accompanying figure.

The widest belt which I have seen in use on a quarter turn was a double belt 12 inches wide. The belt transmitted 60 horsepower.

As to the maximum belt speed consistent with good practice, the following is taken from Kent's "Mechanical Engineers' Pocketbook": "The belt speed for maximum economy should be from 4000 to 4500 feet per minute."

There are often a great many local conditions, however, which prevent the use of this ideal speed, and one must design the belt for the nearest approach to it which may be within the limits of possibility, using the values mentioned as a general guide.

J. A. PRATT.

Williamson School, Penn.

Thermal Storage

In a recent issue, under the caption "Thermal Storage," the results of a test were given, showing that the evaporation had been increased from 123.6 per cent. of the rated capacity of the boilers to 217.3 per cent., an increase of 93.7 per cent., secured by means of so called thermal storage tanks.

It would appear to me much better practice to install live-steam purifiers of generous capacity and thus secure the advantages of the softened water as well as those resulting from the thermal storage.

I desire to score a point in favor of nonautomatic boiler-feed control. Advantages are obtained by maintaining the water level as high as conditions will permit—two and one-half to three gages—the advantages of having enough water stored in the boiler so that the feed pump may be almost closed down for ten or fifteen minutes if necessary, when a peak load comes on, in this way almost doubling the evaporative capacity, thus making it possible to maintain satisfactory service until the firemen have a chance to get the fires up to the usual requirements.

I offer the results of a test of a small light and power plant as an example of what can be accomplished in a small plant during a peak load.

Combined rated capacity of 2 return-tube boilers, horsepower	276
Equivalent evaporation from and at 212 degrees, pounds	14,095
Horsepower developed for one hour	408.6
Per cent. of rated capacity	148
Load in kilowats with two engines	270

The plant was operated with moderate natural draft, and West Virginia coal said to have a heat value of 14,070 B.t.u. per pound was used.

While this load of 270 kilowatts was on, an additional load of 110 kilowatts was added, making it necessary to put on an additional engine. This load of 380 kilowatts was carried for 25 minutes, during which time the feed pump was almost shut down, making it possible to carry the load. This was done for three days in succession, although on the other two days the excess load was slightly lighter. No evaporative test was made while this load was on.

WALDO WEAVER.

Middletown, O.

Combustion and Boiler Efficiency*

By Edward A. Uehling

The importance of carbon dioxide as an index to combustion and, in connection with flue-gas temperatures, to boiler efficiency, is discussed. The curves show the relation between the carbon dioxide and the oxygen.

A steam engine correctly designed, properly constructed, accurately adjusted and adapted to the work to be performed will give maximum economy. The attendant can do little or nothing either to increase or to decrease its efficiency. This is also true of the steam turbine and the dynamo, but the efficiency of a steam generator depends on far more complex conditions, many of which are not only beyond control but are continually changing. Some of these are: first, *atmospheric conditions;* second, *the quality of the fuel,* which at best is never quite uniform, even when similar in kind, and often varies greatly in its content of moisture, fixed carbon, volatile combustible matter, oxygen, ash, etc.; third, *the condition of the boiler* as to setting, dirty heating surfaces, air infiltration, draft, etc; fourth, *the required output,* which varies in some plants between very wide limits.

The heat energy carried off by the flue gas in general boiler-room practice is rarely as low as 15 per cent. and not infrequently reaches 40 per cent. of the calorific value of the fuel burned. This loss is made up of two distinct factors, the sensible heat of the flue gas and the potential heat of the combustible constitutents. Other things being constant, the former varies directly as the stack temperature and inversely as the percentage of CO_2. The latter is independent of the stack temperature, but for any given percentage of combustible constitutents in the flue gas the loss also varies inversely as the percentage of CO_2. The only heat loss that is independent of the percentage of CO_2 in the flue gas is that due to the moisture contained in the fuel, but it is dependent upon the stack temperature.

For any given boiler and setting, driven at a given rate, the temperature of the flue gas depends upon the condition of the heating surfaces and on the amount of air infiltration. When the heating surfaces become dirty, the stack temperature will be correspondingly higher, other conditions being the same. Air filtration lowers the stack temperature in proportion to its ratio to the actual products of combustion. Because of dirty heating surfaces, it should be above normal, but air infiltration may actually bring it below normal, so that the stack temperature alone cannot be depended upon as an index to either economical or wasteful operation, especially since it gives no clue to the efficiency of the furnace. Provided the boiler is properly designed and set and is otherwise in proper condition, its economy depends entirely upon the regulation of the furnace.

Since an excess of oxygen is necessary for complete combustion, it is held by some combustion experts that the percentage of oxygen is a better index to the efficiency of combustion and boiler economy than CO_2. But when it is considered that CO_2 is more readily determined than O, and that there are a number of practicable instruments on the market that will autographically record the percentage of CO_2 in the flue gas at short intervals, there can be little room for doubt as to which of the constituents is the better adapted to serve as the index of economical combustion.

Analyses show conclusively that there is no relation between either O or CO_2 and CO. All that can be discerned with certainty is that there is a tendency toward higher CO as the percentage of CO_2 goes up. The sum of the percentages of CO_2, O and CO contained in flue gas varies with the excess of air, the completeness of combustion, the composition of the fuel and the temperature at which the gas is analyzed. When combustion is complete and the fuel consists of carbon only, the sum of CO_2 and O cannot exceed the percentage of oxygen in the air; that is, 21 per cent. It will always be less than this, because the gas is saturated with moisture, which in general gas analyses is reckoned with the nitrogen. This introduces a possible difference of 1.3 per cent. due to the differences in the moisture content of the gas between 62 and 92 degrees.

When carbon is burned to CO every molecule of O_2 combines with two atoms of carbon and forms two molecules of CO. The gas may, therefore, contain

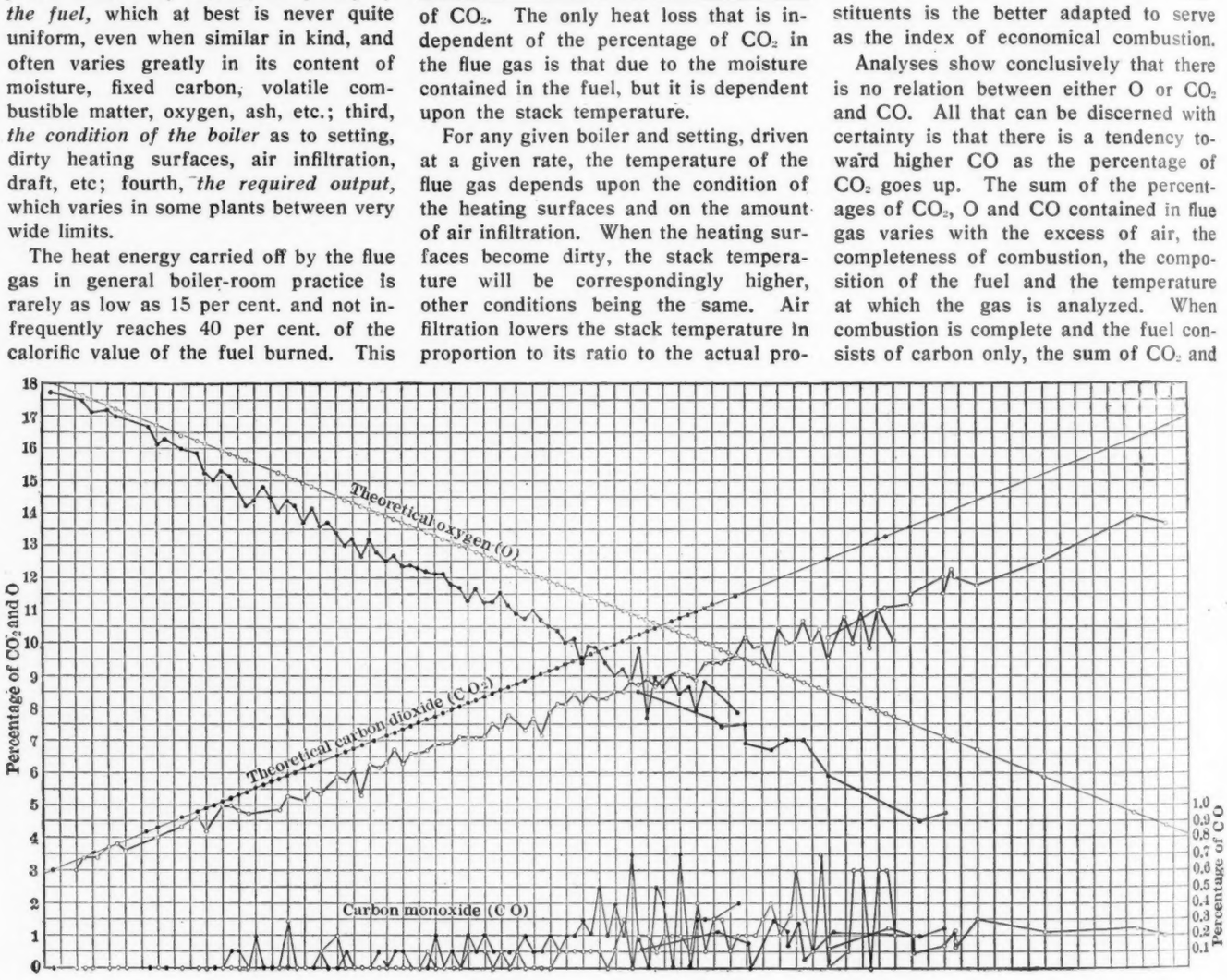


FIG. 1. DIAGRAMMATIC REPRESENTATION OF GOVERNMENT ANALYSIS

*Abstract of paper delivered before the American Society of Mechanical Engineers.

34.7 per cent. of CO as a maximum. If combustion is incomplete and CO is present, the sum of CO_2, O_2 and CO in the dry gas may be greater than 21 per cent. by an amount equal to

$$\frac{34.7 - 21}{34.7} \times CO = 0.4\,CO$$

so that in this case the sum of the oxygen components in saturated gas would be

$$21 + 0.4\,CO - \frac{H_2O \times 21}{100}$$

Since the percentage of CO in normal flue gas reaches 1 per cent. only in exceptional cases and the reduction due to moisture is not likely to be less than 1 per cent., flue-gas analyses in which the sum of the oxygen components exceeds 21 per cent. must be looked on with suspicion. In burning fuel containing a considerable quantity of hydrogen the sum of CO_2, O_2 and CO will always be less than 21 per cent., because all the oxygen which combines with the H_2 forms water vapor, which is condensed, while the nitrogen remains as a diluent.

The highest practicable furnace efficiency is obtained when the greatest possible part of the potential energy contained in the coal is converted into sensible heat by combustion. The highest boiler efficiency results when a minimum of this heat is carried off by the flue gas. The former is realized when combustion is complete, the latter when the weight and temperature of the flue gas are a minimum. Since the weight of flue gas decreases as the percentage of CO_2 increases, high boiler efficiency requires high CO_2. On the other hand, when the furnace is run so as to produce a maximum percentage of CO_2 there is danger of reducing its efficiency because of incomplete combustion due to a minimum of free oxygen. It would seem, therefore, that the highest combined efficiency of boiler and furnace would be obtained when the gain due to reduced weight of flue gas, carried by an increased percentage of CO_2, is balanced by the loss due to incomplete combustion.

Next to careless or unskilful firing, air infiltration causes the greatest loss in the flue gas. Unless especially provided against, it is rarely less than 15 to 20 per cent. of the volume of the flue gas and not infrequently reaches 50 per cent. and over when cracks in the settings are allowed to exist and warped and badly fitting cleaning doors are tolerated.

The curves shown in the accompanying figures were plotted from the results of a number of Government tests. The straight lines represent the theoretical amounts of CO_2 and O which the combustion products would contain if pure carbon were burned with pure air. The analyses were plotted with regard to the percentage of CO_2 and O respectively. The CO_2 was placed in its proper position on the theoretical CO_2 line for the percentages of CO_2 and then the accompanying percentages of O and CO were plotted on the same ordinate in the position called for by their value. In the same manner the O was placed on the theoretical oxygen line and the CO_2 and CO were plotted according to their values on the same ordinate. To show the values of CO more clearly, they are multiplied by five.

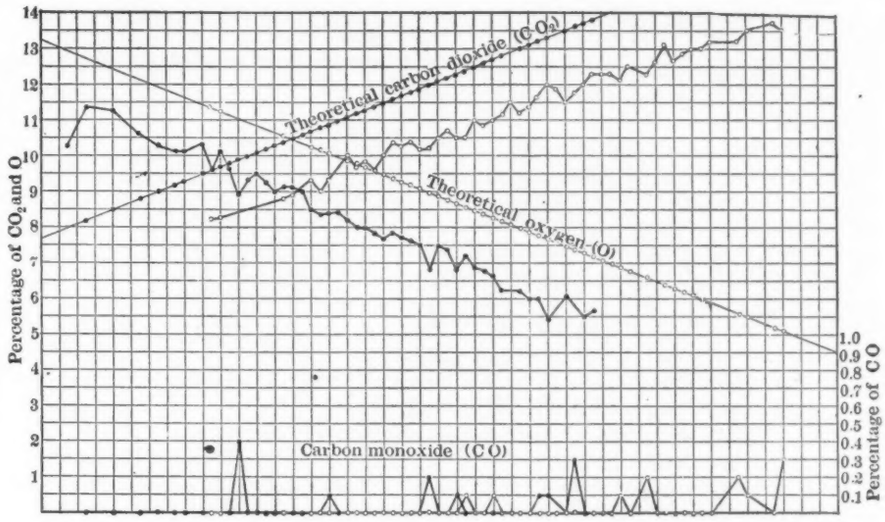


FIG. 2. VARIATIONS OF O, CO AND CO_2 AS SHOWN BY ANOTHER SET OF TESTS

Fig. 1 shows that the percentage of O diminishes as the CO_2 increases, and *vice versa*. It will be noted that both the O and CO_2 lines converge toward the theoretical line as the CO_2 diminishes, which is as it should be if the analyses are correct, because when the CO_2 is zero, the per cent. of O must be that of the air, 21 per cent.

Considering the great variety of fuels represented by the Government analyses it is remarkable with what uniformity the CO_2 and O vary in accordance with what should be expected from theoretical considerations. It is justifiable to conclude that the relation of CO_2 to O in Fig. 1 holds true for all ordinary conditions of simple combustion. It therefore follows that if the percentage of CO_2 contained in the products of combustion is known, the accompanying percentage of O may be inferred from it with a practical degree of approximation. That no such reliable relation exists between the CO and either of the former has already been stated. That this should be so becomes evident when it is considered that the relation between CO_2 and O depends on a natural law, whereas the percentage of CO in a large degree depends on caprice.

The CO line in Fig. 1 illustrates the irregularity of this component. It was stated at the outset that the tendency toward a higher percentage of CO in flue gas becomes greater as the percentage of CO_2 goes up; but the curves show a maximum of over 0.4 per cent. of CO with CO_2 between 10 and 11 per cent., whereas with CO_2 between 11 and 14 per cent. the CO does not average above 0.2 per cent. This apparent anomaly is explained by the fact that the Government analyses includes from 20 to 30 per cent. of infiltrated air and that the real combustion products contained about 25 per cent. more CO_2 and 25 per cent. less O than the analyses show. In reality, therefore, the high CO readings occurred with CO_2 from 12.5 to 13.75 per cent. This would place the high CO in its proper relation to CO_2.

Fig. 2 shows the same characteristics as Fig. 1; that is, the O line follows a trend which should meet the theoretical O line at 21 per cent. when CO_2 reaches zero, and the CO_2 line meets the zero point of the theoretical CO_2 line when O reaches 21 per cent. The CO is irregular and has a moderate tendency toward higher values as the CO_2 reaches the higher values. With the exception of the few analyses previously discussed, this diagram shows very good combustion.

CONCLUSIONS

The conclusions are that the percentage of CO_2 and the stack temperature are the two controlling factors in the production of furnace and boiler efficiency. In view of the fact that both of these components are so easily determined and that apparatus is in existence which will continuously indicate and record them, it is lamentable that more general use is not made of these indexes. It is not generally feasible, nor is it at all necessary, in order to obtain the most economical results, that calculations be based on the ultimate analysis of the coal, but a full analysis of the flue gas should form part of the daily routine. Inasmuch, however, as the most economical percentage of CO_2 varies considerably with the kind of fuel used, and also to some extent on the method of firing, hand or mechanical, the kind of stoker and rate of driving, it becomes

very desirable, if not absolutely necessary, to make a series of tests in which all the essential elements are determined. Having established the most economical value of CO_2, this component of the flue gas will serve as a correct guide to the fireman. An automatic record, preferably a continuous one, for the engineer is of the greatest value. If, in addition, the stack temperature is indicated for the fireman and recorded for the engineer in charge there should be little difficulty in getting maximum efficiency for any particular installation and for any grade of fuel.

Heat Transfer in Surface Condenser*

By George A. Orrok

A series of tests to determine the variation in heat transmission through different condenser materials with variations in the temperature difference, the velocity of the water and the velocity of the steam, and the effect of air mixed with the steam.

Many attempts have been made to ascertain the laws of heat transmission for condenser practice but the results of the various investigators have varied widely. These differences, however, did not trouble designers seriously as long as small condensers and vacuums of over two pounds absolute were the practice, but with the development of the steam turbine the need arose for condensers capable of condensing more than 200,000 pounds of steam per hour at an absolute pressure of less than one pound.

To meet the demand for more definite information upon the subject the writer conducted a series of experiments which confirm and complete much of the work of the earlier investigators. The object in making the tests was to determine the heat transfer through various kinds of condenser tubes and the laws governing its variation under different conditions of steam temperature, pressure and velocity, velocity of circulating water and mean temperature difference.

In order to obtain these data, a small surface condenser was constructed with a relatively small cooling surface and was operated under conditions as near as possible to actual condenser conditions. The steam-inlet, dry-vacuum and circulating-water lines were so arranged and controlled that any desired vacuum, any desired velocity of circulating water and any desired mean temperature difference could be maintained and that any one of these conditions could be varied through a considerable range quite independent of the other two. A hotwell was constructed and so connected to the condenser that the temperature of the condensed steam and its amount could be determined. Thermometers, pressure gages and a water meter were installed and a record was made of all data.

At first steam was taken from a 12-inch exhaust main, but the quality was found to vary so much that it was found necessary to alter the apparatus somewhat and build a small independent boiler for generating steam as it was needed by means of a high-pressure steam coil. The boiler was constructed of two 14-inch flanged tees bolted together, with blank flanges bolted on the ends, and a steam coil, consisting of approximately 22 feet of ¾-inch wrought-steel pipe, was placed in the boiler in such a way that it was entirely below the water level and still left a sufficient disengaging surface for the steam generated in the boiler. Steam from a high-pressure main was supplied to this coil through a 1-inch pipe. Makeup water for the boiler was taken from a city-water line through a ½-inch pipe. A 5-inch pipe connected the boiler with the condenser, which was made of extra-heavy 5-inch wrought-steel pipe with cast-iron screwed flanges on each end; to these were bolted cast-iron blank flanges. A 1-inch condenser tube passed through the center of the condenser and projected through each end, carrying thermometer wells. The bottom of the hotwell was also connected to the boiler by a ¾-inch line containing a gate and a check valve so that the condensed steam could be returned directly by gravity to the boiler; this obviated the necessity of breaking the vacuum and emptying the hotwell after each test. The whole apparatus was as shown in Fig. 1.

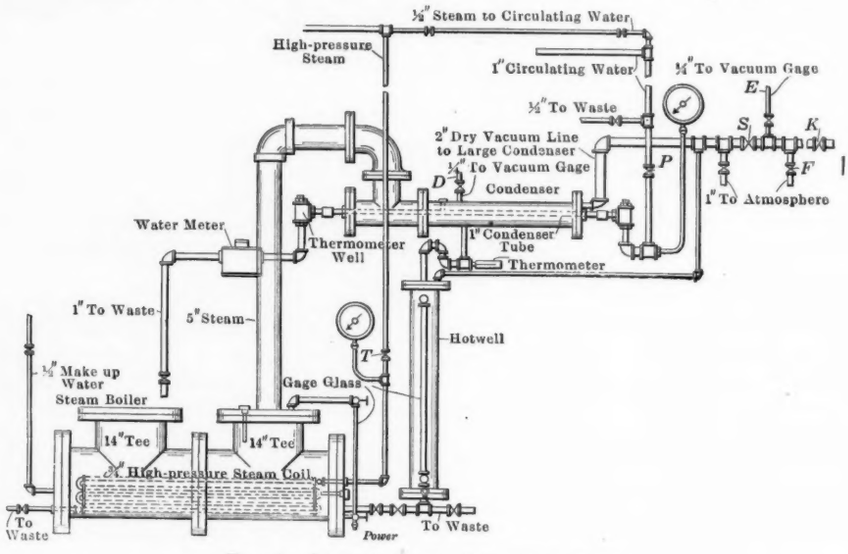


FIG. 1. APPARATUS AS ACTUALLY USED

With this arrangement, for any desired opening of the valve *S*, a constant velocity of steam through the condenser could be maintained by regulating the air valve *F* and the valve *K*, so that the steam passing through the valve *S* had a constant drop in pressure. This drop, due to velocity of the steam, was considered to be the difference in pressure as read at *E* and *D*.

The subject of air leakage was investigated by closing all valves except *S* and *K* in the vacuum line, when it was found possible to hold a vacuum in the small condenser practically equal to that in the large condenser. The valve *S* was then closed and observations were taken as to the rate of fall of the vacuum in the apparatus. It was found that the vacuum would drop only 28 to 27 inches in 15 minutes. These observations were taken before every series of tests.

In order that the condition of the tubes as to cleanliness should not affect the heat temperature, all tubes used for testing were thoroughly cleaned every ten or twelve tests. A series of tests under constant conditions were found to have a maximum variation of not over 10 per cent. and an average variation of 3 to 4 per cent. In these tests, the vacuum was varied by regulating the amount of steam admitted to the steam coil through the valve *T*.

Tests were run at vacuums of 7, 15, 20, 25 and 27 inches of mercury, with the circulating water at a temperature of about 40 degrees Fahrenheit and a velocity of 8.6 feet per second on the following kinds of tubes:

Admiralty; admiralty, oxidized; admiralty, vulcanized on inside only; admiralty, vulcanized on outside only; ad-

*Abstract of paper delivered before the American Society of Mechanical Engineers.

miralty, vulcanized on both sides; admiralty, lead lined; admiralty, old tube from condenser; monel; copper; copper, aluminum lined; aluminum bronze; zinc; tin; cupro-nickel; shelby steel; glass.

Variation of Heat Transfer with Temperature Difference

After the apparatus had been working some time and had been giving consistent results, a set of approximately one hundred tests was run to determine the possible variation of the heat transmission with the mean temperature difference. The quantities of cooling water were so large that the temperature rise was always small and it was considered that the arithmetical mean of the water temperatures and the steam temperature might be used without appreciable error

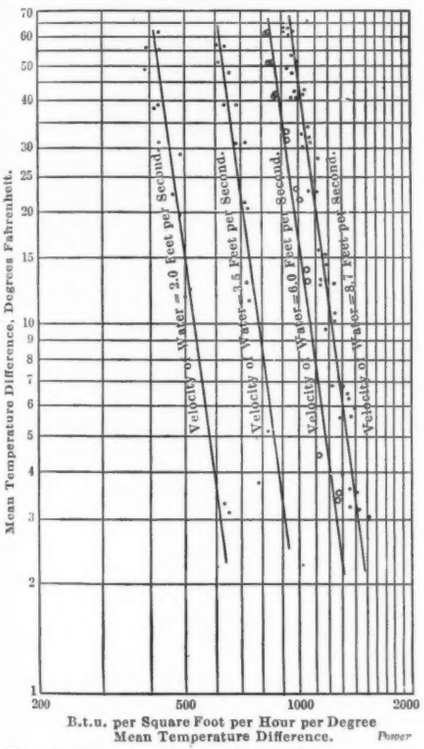

Mean Temperature Difference, Degrees Fahrenheit.
Velocity of Water = 2.0 Feet per Second.
Velocity of Water = 3.5 Feet per Second.
Velocity of Water = 6.0 Feet per Second.
Velocity of Water = 8.0 Feet per Second.
Velocity of Water = 8.1 Feet per Second.
B.t.u. per Square Foot per Hour per Degree Mean Temperature Difference.
Power

Fig. 2. Relation of Heat Transfer to Temperature Difference

in the calculation of the heat transfer. These tests were run at four different velocities of approximately 2, 4, 6 and 8 feet per second, the results were plotted on logarithmic paper and a smooth curve was drawn through each set of plotted points. These curves were practically straight lines, as shown in Fig. 2.

Variation of Heat Transfer with Velocity of Water

A series of one hundred tests were run to determine the variation of the heat transfer to the velocity of the circulating water. These were run at various velocities of from 1 to 11 feet per second and at 15- and 27-inch vacuums, the results of which are plotted in Fig. 3.

Variation of Heat Transfer with Velocity of Steam

That the velocity of the steam across the tube surface has an effect on the transmission has been asserted by a number of authorities. In the present experiments, however, no effect on the heat transmission which might be charged to the velocity of the steam approaching the tube surface was detected.

Consider a condenser consisting of a single tube, through which the cooling water is flowing, surrounded by an atmosphere of steam. Condensation will take place at the outer surface of the tube and the steam will rush in to the tube from all sides. Meanwhile, the space surrounding the tube will attain a constant pressure due to the regular influx of steam into the condenser and its condensation on the tube. This constant pressure will determine the temperature of the steam and if there is no air leakage the temperature will be that due to the saturated vapor of water at the constant pressure. It has been determined experimentally that the temperature and pressure remain sensibly constant throughout the condenser, except for an exceedingly minute space in the neighborhood of the tube surface. There is then a constant flow normal to the surface of the tube, besides such residual velocity as may be in the steam after its passage through the exhaust pipe. This residual velocity in practice is usually from 200 to 600 feet per second and in turbine installations may be as high as 800 feet per second. In direction, it follows the axis of the eduction nozzle until deflected by the tubes, sides of the condenser or baffle plates and constantly loses velocity until, at the further end of the steam travel, this velocity is entirely lost. The component of this residual velocity longitudinally to the tube is of no value in bringing a particle of steam to the condensing surface; neither is the component of this velocity normal to the tube surface, for its action is balanced by carrying away the steam on the opposite side of the tube. Its only action then is to distribute the steam among the tubes.

When a particle of steam comes in contact with the surface of the tube, it is immediately condensed and contracts enormously in volume. At atmospheric pressure the volume of water is approximately 1/1660 of the steam volume. At 28 inches of vacuum, the contraction is about 1/21,500, and at 29.5 inches of vacuum, or ¼ pound absolute, 77,000 cubic feet of steam condenses to 1 cubic foot of water. The vacuum thus created induces a flow of steam normal to the steam surface, the velocity of which, neglecting friction, can easily be calculated for various conditions by means of the formula

$$V = \sqrt{2gh}$$

Transforming this formula,

$$V = 96.25 \sqrt{(P_t - P_a)\,\sigma}$$

where

P_t = Vacuum pressure (pounds absolute);

P_a = Partial pressure due to the air in the condenser;

σ = Specific volume of steam at pressure $(P_t - P_a)$.

The velocity with which the steam

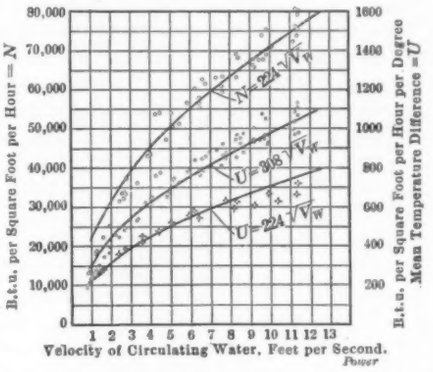

B.t.u. per Square Foot per Hour = N
B.t.u. per Square Foot per Hour per Degree Mean Temperature Difference = U
N = 224 √Vw
U = 368 √Vw
U = 224 √Vw
Velocity of Circulating Water, Feet per Second.
Power

Fig. 3. Variation of Heat Transmission with Water Velocity

particles strike the tube surface is apparently independent of the distance they travel before striking and is reasonably independent of the vacuum. The velocity varies from about 1750 feet per second at 1 pound absolute to about 1900 feet per second at 14.7 pounds absolute. These values of V are shown in Fig. 4 and the effect of a small quantity of air in the condenser on the velocity of flow of the steam is shown to be very small.

The Effect of Air upon Heat Transmission

During the early part of these tests it was impossible even to approximate air-free steam, so that consistent transfer results could not be obtained. With

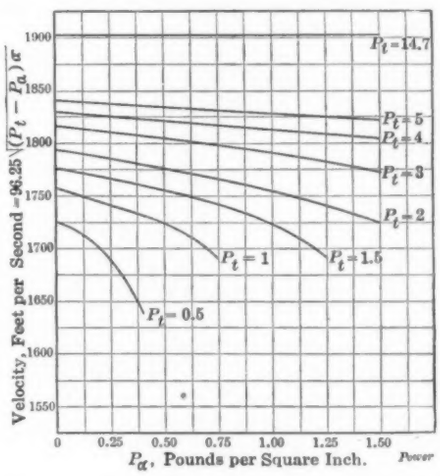


Fig. 4. Variation of Heat Transfer with Steam Velocity

the changing of the apparatus, however, it was possible to keep the air down to a rather small quantity and the results obtained were much more consistent. It is probable in these cases that the partial air pressure never exceeded ½ inch and in most cases was below this. By close

attention to all possible sources of air leaks in the final tests, it is probable that the partial air pressure never exceeded ¼ inch and was sometimes much less than this.

Conclusions

The heat transferred from steam surrounding a metallic tube to cold water flowing through the tube is proportional to the seven-eighths power of the mean temperature difference of the water and steam temperatures. This is equivalent to the statement that the coefficient of heat transfer is inversely proportional to the eighth root of the mean temperature difference.

The coefficient of heat transmission is approximately proportional to the square root of the velocity of the cooling water.

This coefficient is independent of the vacuum and of the velocity of the steam among the tubes or in the condenser passages. It may be proportional to the square root of the velocity normal to the tubes, but in all common cases this velocity does not vary more than a tenth part.

The effect of air upon the amount of heat transferred is very marked, particularly at high vacuums, and most of this air is due to leakage through the walls and joints of the apparatus.

Taking the heat transfer of the copper tubing as 1, under similar conditions the transfer for other materials is approximately as follows: admiralty, 0.98; aluminum lined, 0.97; admiralty oxidized (black), 0.92; aluminum bronze, 0.87; cupro-nickel, 0.80; tin, 0.79; admiralty lead-lined, 0.79; zinc, 0.75; monel metal, 0.74; shelby steel, 0.63; old admiralty (badly corroded), 0.55; admiralty vulcanized inside, 0.47; glass, 0.25; admiralty vulcanized on both sides, 0.17. This coefficient (due to the material of the tube) will be designated by μ. Corrosion, oxidation, vulcanizing, pitting, etc., have also a marked effect in reducing the transfer. This reduction, best shown by the admiralty tube which gave $\mu = 0.55$, may reduce the transfer at least 50 per cent. The foregoing conclusions may be expressed mathematically as follows:

$$U = K\frac{C\rho^5\mu\sqrt{V_w}}{\theta^{\frac{1}{8}}}$$

where

U = Coefficient of heat transmission;
C = Cleanliness coefficient varying from 1 to 0.5;
μ = Material coefficient varying from 1 to 0.17;
ρ = Ratio of the partial pressure due to the steam, to the total steam and air pressure, which varies from 1 to 0;
V_w = Water velocity in feet per second;
θ = Mean temperature difference;
K = A constant, probably about 630.

The effect of the length of tube, or rather, the length of water travel, has not been considered and the design of the condenser must be such that there is a free steam passage to every tube.

This expression for U is cumbersome to use and for modern turbine condenser work certain conditions may be taken as well settled. The guaranteed vacuum is usually 28 inches; the entering circulating water is usually 70 degrees and a 20-degree temperature rise is considered economical. Under these conditions $\theta = 18.3$ and $\theta^{\frac{1}{8}} = 1.44$, θ being calculated on the geometrical curve as 18.2. For these cases it will be nearly as accurate and much simpler to calculate θ by the logarithmic method, neglecting θ in the denominator and using 435 or $\frac{630}{1.44}$ for K^1. The expression will then be

$$U = K^1 C P^5 \mu \sqrt{V_w}$$

This equation agrees with the results of a number of tests on full-sized condensers under varying conditions. There were no attempts to determine the amount of air handled by the air pump in these cases, but the amounts of air indicated by the formula are such as agree with the pressures and temperatures taken.

Test of a 9000 Kilowatt Turbine*

By F. H. Varney

Late in December, 1908, the Pacific Gas and Electric Company installed a 9000-kilowatt vertical turbo-generator set to operate in connection with its other stations in Oakland, consisting of one gas-engine, three steam-engine and eleven water-power plants. Beside the demand for power which the set was expected to meet, it was also thought that the regulation would be improved, which expectation was realized.

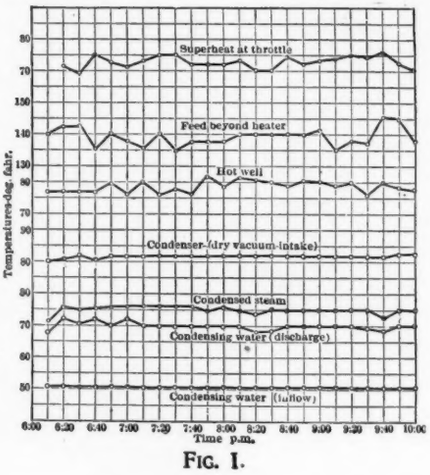


Fig. 1.

Owing to the urgent need for the plant, the efficiency test was postponed until the station had been in operation for 14 months. The conditions of the test were as follows: The turbo-generator was run on a closely regulated commercial load, which was maintained as nearly uniform as possible by the load despatcher. Two runs of four hours each were made at loads approximating 7000 kilowatts and 9000 kilowatts.

As the tests, two in number, were conducted for the purpose of determining only the efficiency of the turbine, provision was made for the observation and measurement of the water from the turbine, the total make-up and auxiliary water, the power delivered by the generator at the switchboard, the exciter output and the pressures and temperatures required to complete and interpret the observations.

The various data were plotted and several charts constructed of which Figs. 1 and 2 are charts plotted from the data obtained in the second test.

The contract specifications provided for a steam pressure at the throttle of 175 pounds gage, a superheat of 125 degrees Fahrenheit and a vacuum of 28 inches.

The guarantees were as follows:

Generator Output, Kilowatts.	Pounds Water per Kilowatt-hour.
5000	16.75
7000	16.40
9000	16.40

Variations from the specified conditions were to be subject to the following corrections:

For each 10 pounds pressure.....1 per cent.
For each 12½ pounds deg. superheat.1 per cent.
For each inch vacuum....1.08 lb. per kw.-hr.

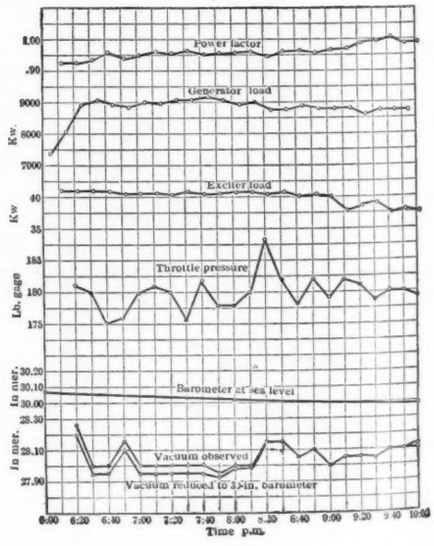


Fig. 2.

In the second test, lasting from 6 p.m. to 10 p.m., the final corrected values gave a water rate of 15.42 pounds per kilowatt-hour.

*An abstract of a paper read before the American Society of Mechanical Engineers.

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POWER and The Engineer

Editorial

The Work of the Institute

In the prospectus of the Institute of Operating Engineers attention is called to the lack of unity of purpose which results in an unsettled condition and weakness in a vocation followed by more than a quarter of a million of men in the United States. It is to assist in placing the vocation of steam engineering on the high plane of honored and respected callings which, though not wholly professional, require professional attainments of a high order if mastership is desired.

While palliative measures are discussed and ameliorative conditions sought for the various classes of employees, the operating engineer, the nature of whose work is largely solitary, is being forgotten. There is no systematic method of advancement where engineers are employed in numbers nor is there any accepted standard of proof of merit for the man who has mastered his vocation step by step and is able and ready to demonstrate his fitness to fill the position of highest responsibility in any power plant. There is, on the other hand, a constantly increasing difficulty in obtaining really skilled operating men; men with manual training, mental equipment, demeanor, tact and personal appearance fit for the positions to be filled. This difficulty is in part, if not wholly, caused by the lack of any effort to secure coöperation between the employer, the operating engineer and the ambitious subordinate worker. Sooner or later the economic relation between these factors, the bread and butter question, must be recognized, met and settled. In the methods proposed by the Institute of Operating Engineers an effort is initiated which it is expected will meet this question in a broad, clearly defined and systematic manner.

The institute itself is to be a nation-wide coöperative college; and as the time has passed when a college was constituted by merely a group of buildings, a campus and a corps of instructors, the students can pursue their studies at home, at the evening school or where they will. Their correspondence with the home office and their examination papers will give their standing and mark their attainments.

Already many employers of skilled labor in many lines have recognized the importance of this new movement and have signified their appreciation by offers of coöperation with, and financial support, of the attempt to create a supply of trained men so associated that determination to be proficient in a chosen calling is the mainspring of action.

The final results achieved by the institute will be a unity of purpose to place the vocation of steam-engine operation on the high plane of skilled pursuits to which it is entitled by its importance.

Boiler Inspections and License Laws

The boiler inspector's duties are very exacting and he does not receive the hearty support and coöperation from a great many of those with whom he is brought in daily contact. When an inspector sends out a notice that he will be at a certain plant to make an internal inspection of a boiler, say, at 8:30 on a Sunday morning, and requests that the boiler be gotten ready for the inspection, the chances are that the work will not be half done. The inspector has to put up with these conditions in every State in the Union, except Massachusetts, in which State he can decline to make the inspection if the boiler is not properly prepared and will not issue a certificate until an efficient inspection has been made.

In many cases the engineer is not to blame, as frequently the notice is not sent to him personally, but to his employers and they do not notify him until it is too late to get the boiler in proper condition. In fact, sometimes they do not notify the engineer at all, and when the inspector arrives at the plant, he finds it closed, or in charge of a watchman, but no boilers ready. Sunday being a busy day for the inspector, he leaves his card with the watchman to give to the engineer or tacks it on the engine-room door, and visits the next plant on his list, where he is very likely to meet with the same reception.

Again, the inspector often finds it necessary to have repairs made, but the engineer, in many cases, has not the authority to order them and has to refer the matter to his employers, in which case the employers often consider the repairs unnecessary and will delay having them made until an accident occurs. In cases like this an inspector's life is not a happy one, for he is between two fires; he knows the repairs are actually necessary but has not the power or au-

thority to compel the shutting down of the boiler until they are made, and if he suspends the insurance until such repairs are made, his company will probably lose all the insurance that the owners of the plant carry with them. However, the inspector should not take any risks, for by suspending the insurance he is not only protecting his company but is protecting the owners of the plant and their employees, besides his own reputation.

Every engineer should know how and should in every way assist the inspector to make a thorough examination of the boilers, and, if repairs are recommended, should have them made at once. It is an old proverb and a good one that says, "A stitch in time saves nine." In Massachusetts the engineer's and the inspector's duties are made plain by the boiler-inspection law, and it would be advisable for every man in charge of a steam plant to get a copy of that law and study it carefully.

The coöperation of all engineering organizations, and the support of all engineers who are not members of such organizations are needed to use their influence to put into effect proper license laws. If the National Association of Stationary Engineers would hold biennial sessions instead of annual sessions and devote a sum of money annually toward helping to procure license laws, it would not be many years before every State would enact laws that would prove a benefit to the owners of power plants and their operatives. The question of wages would take care of itself, for the reason that the engineer would then educate himself so as to entitle him to the highest grade of license, and not merely be the "stopper and starter" so often referred to. By such actions the national association would be living up to the full text of its preamble, which reads: "Its meetings shall be devoted to the business of the association and at all times preference shall be given to the education of engineers, and to securing the enactment of engineers' license laws in order to prevent the destruction of life and property in the generation and transmission of steam as a motive power." Nothing can be plainer than this, and the subordinate associations should demand aid and assistance from the national organization in procuring license laws.

The engineer is the most important employee in a power plant, and he should thoroughly understand his plant from A to Z. Many men now operating steam plants know nothing about the construction of the steam boilers they are operating, the thickness of the shell, thickness of the heads, the strength of the seams and the number and location of the braces; and, not knowing these few points, they cannot tell whether or not they are carrying too much pressure on the boilers, but simply carry the pressure because it is necessary for the operation of their plant. If New York and other States had laws similar to those of Massachusetts and these were properly enforced, life and property would be better protected and the engineers who are now opposed to license laws would in a short time be their strongest advocates. State Senator George L. Meade in his speech at the opening of the national convention in Rochester last September stated that a competent licensed engineer is a far better protection against boiler explosions than any system of inspections that could be devised, for the reason that he makes it a point to thoroughly examine his own boilers; also, that the granting of licenses in all the professions and trades had elevated the standard of those licensed. When we have such encouraging words from a State senator, the engineers of New York State should make an effort to have the incoming legislature enact the proper laws.

A Shaftless Pump

Like all new contrivances of simple construction, the Humphrey pump incites wonder as to why "somebody" hadn't thought of it before. This apparatus is an internal-combustion pump in which the water itself serves as a piston and its oscillations, due to kinetic energy delivered to it by the explosion of the charge, are utilized to expel the burned gases, draw in the fresh charge and compress it for the next explosion. It has no shaft or other mechanical parts, excepting a few simple valves. In this issue appears a part of a paper recently read by the inventor of this ingenious apparatus in which some important modifications are described. It is rather surprising to learn that the pump can be built to lift water from a level below the intake valves and deliver it against pretty high pressures. The modification of the simple pump by means of which this is feasible is not included in this week's section of Mr. Humphrey's paper because of space limitations; it will be described next week.

An even more surprising possibility of the pump is the alteration of the compression pressure without changing the hight to which the water is raised. In the simple pump, the hight to which the water is driven by the explosion of the charge determines the pressure to which it will compress the next charge on its second return "stroke." By merely adding a bell, projecting into the combustion chamber, the compression space, outside the bell, may be proportioned to give high compression without high cushion pressure. This arrangement also effects complete scavenging of the space into which the fresh charge is drawn.

The "combinations" of which the underlying principles of the pump are susceptible without entailing complicated construction appear to be extraordinarily numerous.

Inches or Pounds

At this time when everything relating to the generation of power and its transmission is moving or has only recently emerged from a state of transition from one base or form of nomenclature to another, it would seem opportune to establish a correct method of using some few engineering terms. If a navigator should use miles in designating distances eastward and degrees of longitude when referring to a westward direction, he would present no more incongruous spectacle than the engineer who measures pressures up to that of the atmosphere in inches of mercury reckoned backward and thence in pounds, calling that pressure one pound which is really 15.7 pounds per square inch.

Taking Unfair Advantage

When an engine meets with an accident unscrupulous builders of competing machines lose no time and spend no pains to exploit the misfortune to the discredit of the engine, and to prejudice intending purchasers against it, and this whether the engine was at fault or not. An instance of this kind was the breaking of a receiver on a McIntosh & Seymour engine at the Harvard station of the Boston Elevated Railway Company. The accident was a plain case of overpressure, as was plainly stated in our article describing it. The subjection of this receiver to a pressure several times that which it was designed to bear, was no fault of the engine or its builders. Of this we have the positive assurance of the company's engineers. Any receiver, unless of abnormal proportions, would have failed under the conditions to which this one was subjected and nothing that the engine builders could have foreseen or provided would have prevented it. Any citation of the accident to the prejudice of the engine is unfair and uncalled for.

Smoke Up and Deduct the Fine from the Coal Bill

Such an absurd ruling was recently made by Judge Hines in the Municipal Court of Chicago. It appears that the Boston Fuel Company sued the Cooke Brewing Company for $999. The company last named had been fined $100 for violating the smoke ordinance, and deducted that amount of money from its coal bill. The Boston Fuel Company, of course, sued to recover and the jury determined that if coal produced objectionable black smoke and if the user of that coal were fined as a consequence, the amount of the fine could be deducted from the coal bill. The suit will be appealed and it is to be hoped that some of the smoke in question does not befog the minds of those who will render the verdict.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Proportions of Hose Nozzles

What are the best proportions for a hose nozzle to throw a stream the farthest?

P. H. N.

In common practice, nozzles are made 6 to 10 times the hose diameter in length with the hole at the end one-third the hose diameter.

Water Leg of a Boiler

What is meant by the term "water leg" and what part of a boiler is it?

W. L. B.

The term applies to the narrow water space between the inner and outer sheets of the firebox of vertical and locomotive types of boilers.

Steam for Preventing Clinkers

I have been told that a jet of steam under the grates will prevent the formation of clinkers. Is this true, and is it an economical practice?

S. P. C.

Clinkers are caused by the melting and running together of the incombustible in the coal by the heat of the fire. If steam enough is passed through the fire to keep the temperature below the melting point of the ash, clinkers will not form. It is certain that there is no economy in using steam to reduce the temperature of the fire under the boiler which makes the steam.

Horsepower and Boilers

What is the correct number of square feet of heating surface to be reckoned for a horsepower in different types of boilers?

H. P. B.

Ten square feet of heating surface per horsepower has always been the standard used by the makers of water-tube boilers, and from 12 to 15 the standard of the makers of other types. But recently the opinion has prevailed that the heating surface in other types is as efficient as that in the water tube and 10 square feet of heating surface per horsepower is now the rating for all types of boilers.

Two Hundred-Horsepower Horizontal Boiler

Are horizontal return-tubular boilers of 200 or more horsepower in successful use anywhere, and what would be the dimensions of one of 200 horsepower?

H. H. B.

They are made and used up to 300 horsepower. One of 200 horsepower would be 7 feet in diameter, containing one hundred and twenty-four 3½-inch tubes, 18 feet long, and for 125 pounds pressure be made of ⅝-inch plates with butt and strap joints quadruple riveted.

Diameter of Air Compressor Valves

I find a difficulty in determining the proper diameters of air-compressor valves of the poppet type and should be obliged if you would give me what is considered good practice. Some say the velocity of air through the valves should not exceed 2000 to 3000 feet per minute; others say that 5000 to 6000 can be taken with safety.

A. C. V.

Modern practice for piston speeds of 500 to 700 feet per minute dictates that the opening through the discharge valve shall be from 15 to 25 per cent. of the piston area. Inlet-valve passages should be as great if possible. While the volume of air after compression is less than at admission the time of discharge is much less than that of intake, causing an increase in speed of discharge. It is not necessary that the peripheral area under the lip of the open poppet valve should be as great as the port area through the seat, as the restriction at valve lip seems to be so momentary that full lift is not required.

External Corrosion of Boilers

What are the most common causes of external corrosion of boilers?

E. C. B.

Leaking seams, rivets and gaskets, also wet ashes and soot if permitted to remain in contact with the plate.

Underground Pipe Covering

I have two lines of steam pipe each 400 feet long laid underground in a wooden box packed with shavings. This method is unsatisfactory for the shavings gather moisture and the pipes corrode on the outside. How would it do to lay them in tile?

W. P. C.

The same care should be exercised in laying underground steam pipes as where run overhead. They must be well insulated and kept dry. A perfectly drained trench with the pipes run above the possibility of contact with water only will give satisfactory results.

Steam per Horsepower Hour

What is the average number of pounds of steam required per horsepower per hour by simple automatic cutoff engines of 100 horsepower and by compound-condensing engines of 500 horsepower?

S. H. H.

For the simple engine 25 to 30 pounds, and for the compound-condensing about one-half as much.

Temperature of Red Hot Iron

What is the temperature of iron at a dark red heat?

T. H. I.

Iron shows red at a temperature of 700 degrees in the dark; in daylight at 1000 to 1300 degrees, according to the light.

Steam Pipes and Mains

I have two 17x22-inch engines running 175 revolutions per minute. What should be the size of the steam pipes and the main which supplies them?

S. P. M.

A formula for finding the diameter of the steam pipe for an engine reads:

$$Diameter = \sqrt{\frac{HP}{6}}$$

A 17-inch engine should develop 175 horsepower and substituting this value in the equation, it reads:

$$\sqrt{\frac{175}{6}} = 5.4 \text{ inches}$$

the diameter of a pipe necessary to supply one engine. To supply two engines the pipe must have a carrying capacity of two pipes, each 5.4 inches in diameter. To find the diameter of one pipe which with the same head will have a carrying capacity equal to two smaller ones of the same size, multiply the diameter of the smaller by 1.3;

$$5.4 \times 1.3 = 7.02 \text{ inches}$$

the diameter of steam pipe required to supply two 17-inch engines. In selecting the pipes the nearest commercial size should be taken.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Impulse Reaction Turbine

Messrs. Willans & Robinson, Ltd., and E. G. Izod, of Victoria Works, Rugby, England, have recently evolved and patented a combined impulse and reaction turbine which can be operated with either low-pressure steam, high-pressure steam or a combination of both low- and high-pressure steam.

The following information was obtained from a description in a recent number of *The Mechanical Engineer,* London:

The impulse wheel is made of larger diameter than the end of the drum or reaction portion of the turbine which is adjacent to it, and the portion of the outer casing within which it is inclosed is similarly made of larger diameter than the portion of the casing which surrounds this end of the drum. The high-pressure nozzles for delivering steam to the impulse blades pass through one side of the enlarged portion of the casing which surrounds the outer portion of this wheel, while an inlet for low-pressure steam is made through the opposite side of this portion of the casing. The steam passage leading to this low-pressure steam inlet is led back alongside of the high-pressure steam passage, and a stop valve is so arranged in each that both may be operated by the movement of one lever extending from the main governor of the turbine. Stop valves in the high-pressure and low-pressure steam passages are also provided to allow of the turbine being operated either by high-pressure steam alone, or by low-pressure steam alone, or by a combination of high- and low-pressure steam.

Referring to the plan shown in Fig. 1, *A* is the reaction drum, *B* the wheel of larger diameter, *C* the fixed outer casing with the end at which the wheel is situated correspondingly enlarged in diameter, *D* a nozzle passed through one side of the enlarged portion of the casing for directing high-pressure steam against the blades of the impulse wheel, *E* a port formed through the opposite side of this enlarged part of the casing for admitting steam of low pressure to the space between the impulse wheel and reaction drum, *F* a valve on the high-pressure steam-supply pipe, *G* a valve on the low-pressure steam-supply pipe and *H* a lever operated by a governor at *I*, by which movement may be given simultaneously to both valves.

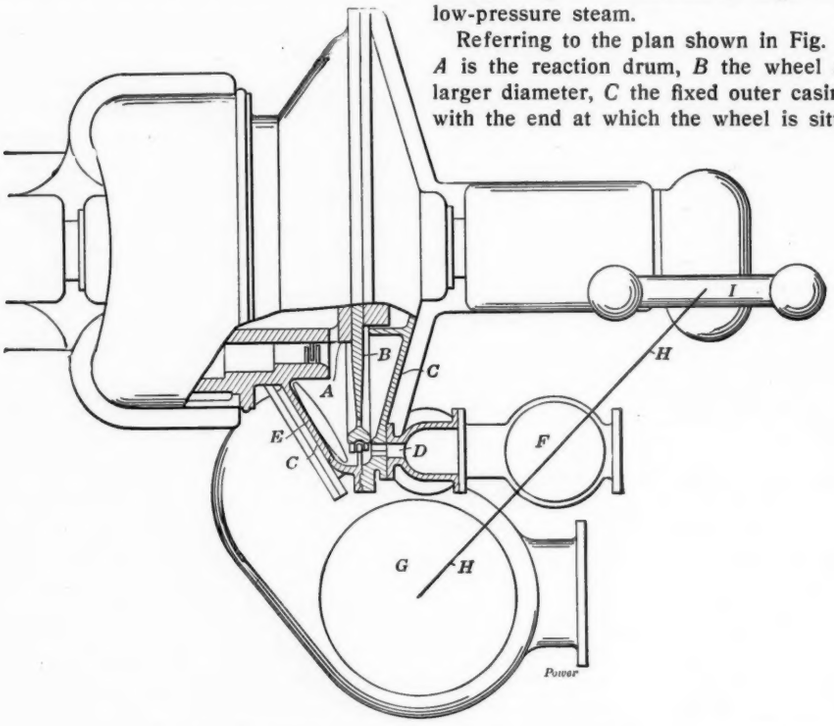


FIG. 1. COMBINED IMPULSE AND REACTION TURBINE

The sectional elevation, Fig. 2, shows the means employed for operating the two valves from the governor lever *H*. This lever is mounted on a fulcrum *J*, and from one end of it the low-pressure valve *G* is acted on as shown, while the high-pressure valve *F* coupled to a rod *K* is connected to lever *H* by means of a regulating lever *L* mounted on fulcrum *M*, the motion from the main lever *H* being imparted by means of stops *N* and *O* which are fixed to lever *H*, the top stop *O* being adjustable.

The spring *P* bearing against a collar on rod *K* and pressing against spring case *Q* always tends to close the valve *F*. The spring *R* being a compensating spring allows the low-pressure valve *G* to have a slight lead, and enables both valves to be tightly closed on their respective seats.

By regulating the adjustment *S* on the low-pressure gear and the stop *O* on the high-pressure gear, it is possible to run the turbine under the following conditions: As a high- or low-pressure turbine; or as a mixed pressure; and the high- or low-pressure steam may be supplied or cut off at any given predetermined load by means of these adjustments.

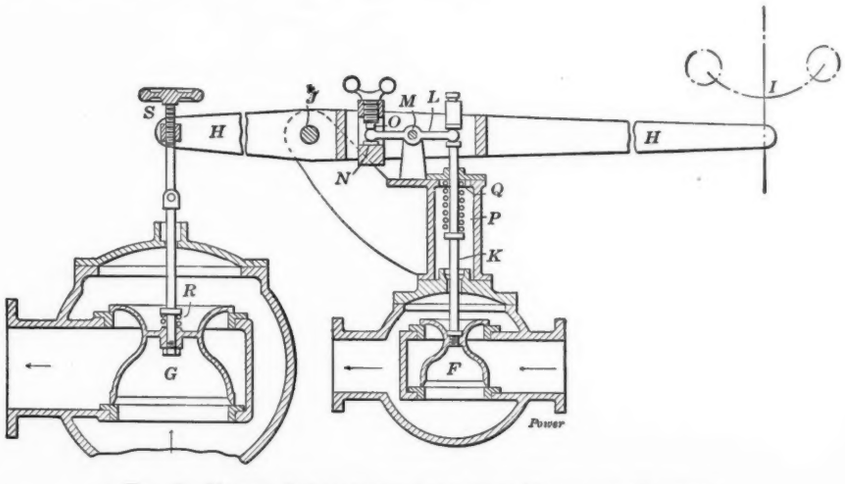


FIG. 2. VALVE ARRANGEMENT OF THE COMBINED TURBINE

The Heath Valvograph

W. F. Stanley & Co., Ltd., of Great Turnstile, London, have recently placed on the market a simple and accurate instrument by means of which it is possible to read from, or construct, valve diagrams, and solve with the least possible construction and without calculation the various problems which arise in designing a new valve or when making alterations to an existing one. It is adaptable to all valves where the Zeuner diagram can be used, and when once the principle of the instrument is understood, anyone may become proficient in its use with very little practice. In the hands of an experienced engineer it will also be found most useful and time saving, while to the student it will afford great assistance, enabling him to study more closely than he could by any other means the effects produced by modification of the steam distribution.

The instrument, according to *The Mechanical Engineer,* of London, is made in transparent celluloid, and is marked as

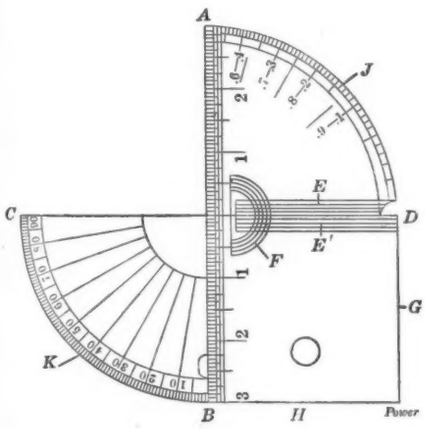


FIG. 1. HEATH'S VALVOGRAPH

shown in Fig. 1 with the following: An advance line *A B* giving position of the eccentric, and also readings for lap and port opening; a cutoff and admission line *C D;* and lead lines *E*. These are four in number and give lead by sixteenths up to ¼ inch; the lines *E'* and *F* are for use when determining the valve travel. Divisions *K* form an ordinary protractor for use when measuring the angle of advance; and a special protractor *J*, divided to give cutoffs in fractions of the stroke, is also given.

To use this when the cutoff falls on the right-hand side of the diagram, the instrument is placed in a vertical position with its axes *A B* and *C D* coinciding with those of the diagram, when a line drawn from the center through the required cutoff on the protractor will intersect the valve-travel circle at the correct point. If, however, the cutoff should fall on the left-hand side of the diagram, the instrument is moved round so that the required cutoff on the protractor coincides with the horizontal axis of the diagram; then a line drawn from the center to *C* will give the cutoff as before. To aid rapid setting, gage lines are marked on this protractor giving a cutoff advancing by eighths in place of hundredths, and two square edges *G* and *H* allow the instrument to be moved to any part of the diagram without alteration of the angle at which it has been set.

Referring to Fig. 1, it will be seen that the instrument is divided to read lap, lead and port opening to 1/16 inch, point of cutoff to one-hundredth of the stroke and angle of advance to one degree; but, when required, finer readings than these may be made by estimation.

By placing the instrument over any valve diagram, as indicated in Fig. 2, it will be noticed that the advance line *A B* coincides with the angle of advance, the cutoff line *C D* intersects the valve-travel circle at points of cutoff and admission, and is tangent to the lap circle, the lap and port opening being read on the scales marked on the advance line, while the lead (if any) is indicated by the particular lead line *E*, which cuts the valve-travel circle at the point where it is intersected by its horizontal axis. From this it is apparent that in whatever position the instrument may be placed over a circle which represents the valve travel, provided *A B* passes through the center of that circle, lap, port opening, angle of advance, points of cutoff and admission and lead will always remain in correct relation to one another. Thus, in setting down a diagram, the rectangular axes *L M* and *N O* are first drawn, and then with *X* as center a circle having a radius equal to half the valve travel is described. The instrument is now placed over this figure so that it registers any two known dimensions, when it will be set for all other readings. While thus set, with a finely pointed pencil, lines *A B* and *C D* are drawn, when the diagram will be complete for the steam side, and all readings may be taken from it by means of the scales and protractors provided. When the cutoff is known, this should first be marked off as already described, and used as one of the points for setting.

To construct the exhaust side of the diagram, it is only necessary to place the instrument so that *A B* falls on the angle of advance and the zero point of the two scales marked on the advance line falls below the center *X* to the extent of the exhaust or inside lap, if any; then *C D* will cut the valve-travel circle at the points of release and compression. Line *C D* is then drawn in as before.

In cases where the valve travel is unknown, but the cutoff lead and port opening are given, the travel may be determined in the following manner: The axes *L M* and *N O* are drawn, and with *X* as center a circle is described (lightly), having a radius equal to the port opening; also, by aid of the cutoff protractor, the cutoff line *X P* is set up. The instrument is now placed in such a manner that the smallest arc of the series *F* touches the line *X P* and the arc which is a distance from the smallest one equal to the desired amount of lead touches the horizontal axis *L M*, the line *C D* being within the angle *P X L*, and the center *X* being cut by the line *E'*, which equals half the lead; that is, if the lead happened to be ⅛ inch, the arc which incloses two spaces of ⅛ inch would be used in conjunction with the line which is one space or $\frac{1}{16}$ below *C D*. A line is now marked along the exposed portion of *C D*, and the instrument is then moved by aid of the square edge *H* till *A B* is tangent to the port-opening circle. With the exposed portion of *A B* this tangent is drawn to cut the line *X P*, and the angle so formed is bisected by letting any one of the arcs *F* touch these two lines while *C D* passes through the point where they intersect. A

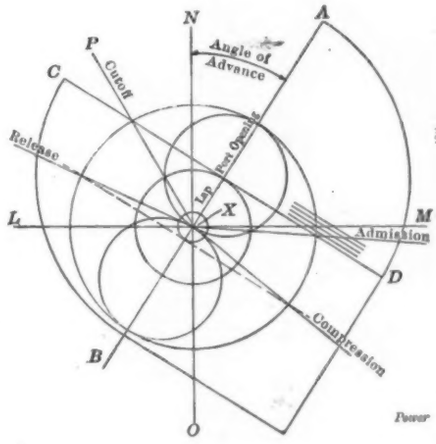


FIG. 2. INSTRUMENT PLACED OVER VALVE DIAGRAM

line is now drawn in as before, with the exposed portion of *C D*, and the point where it cuts the first line will be a distance from the center *X* equal to half the valve travel. The valve-travel circle is then drawn with this radius, and the diagram completed as already described.

When the valve travel exceeds 6 inches, the diagram should be drawn half size, and all readings, except those of the two protractors, doubled. Should the diagram be of the opposite hand to that shown in Fig. 2 the instrument is inverted, and used as already described, only using lines *E'* as lead lines and line *E* for determination of valve travel. It is always a good plan to set down the diagram on some corner of the valve drawing, as it forms a handy record which may be consulted at any future time, and when desired the lap and valve circles may be added, and the diagram made quite complete.

Automatic Water Jet Eductor

The Koerting automatic water-jet eductor is used for automatically pumping water. A sectional view of the eductor proper is shown in Fig. 1. It operates as follows:

A jet of pressure water passing from a narrow into a wider nozzle thereby creates a vacuum. This causes the water to be transported to rise in the supply pipe and to pass mixed with the pressure water through the wider nozzle to the discharge.

The installation of this eductor simply consists in making a few pipe connections, and in starting the motive power, i.e., the pressure water. The pressure in pounds per square inch at the eductor

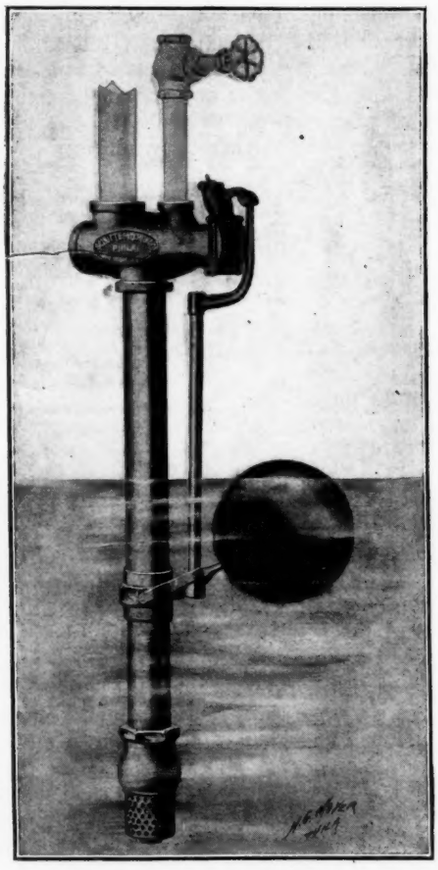


FIG. 1. AUTOMATIC JET EDUCTOR

should be 2½ times the elevation in feet. So, for instance, a pressure of 25 pounds is desirable for an elevation of 10 feet. The eductor takes water of any temperature up to the boiling point, and is built for an hourly capacity of from 125 to 20,000 gallons.

The water-jet eductor is used for removing water from pits and quarries, for emptying foundation pits and gas-holder tanks, for quenching coke, for discharging ashes from stationary or marine boiler rooms to a convenient location, for washing the filter sand in water works and to sink mining shafts.

The special feature of the eductor, Fig. 2, is the float which starts the operation automatically whenever there is water in the pit to be emptied, turning on the pressure water full, and stopping as soon as all the water is removed. The eductor is installed in a wooden or cement barrel. It is made by the Schutte & Koerting Company, Thompson and Twelfth streets, Philadelphia, Penn.

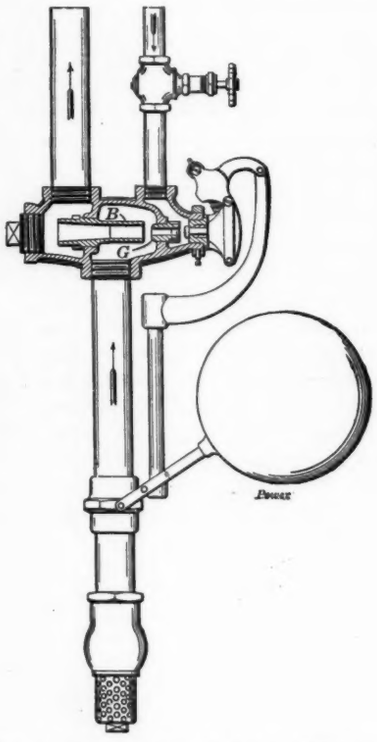


FIG. 2. SECTIONAL VIEW OF OPERATING MECHANISM

Sarco Steam Trap

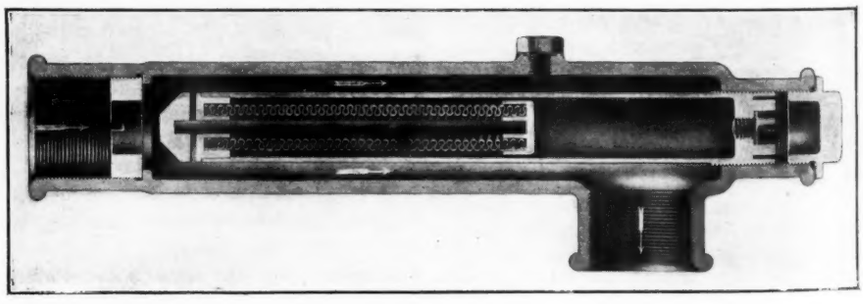
The apparatus illustrated herewith consists of a valve with a conical top, which derives its motion from the expansion of a fluid that is extremely sensitive to fluctuations in temperature.

The fluid is contained in a hermetically sealed tube and the whole fitted to a casing suitable for direct connection into the steam line. While this is essentially a low-pressure trap, it can be used for high-pressure work. The steam, however, must flow in the direction of the arrow. For low pressures it makes no difference in what direction the steam passes through the trap, although it is better to have the flow in the same direction as for high-pressure work.

The trap can be installed in any position desired. It is easy to clean without disconnecting by simply blowing steam throught it. There can be no freezing of the steam pipes as the trap opens automatically with the cooling of condensation and stays open until steam enters the trap, when the valve closes. All condensation is ejected as fast as it forms so that the line is kept clear. The inner cartridge can be replaced at small cost, should it deteriorate, and without taking the trap off the line.

To place the trap in operation it is necessary to connect it with a pipe and then unscrew the cartridge or inner tube about ¾ inch and allow steam to flow through freely for about one minute. The expansion of the chemical fluid in the cartridge will cause the piston to advance and rest against the valve seat. After this the cartridge is screwed up against the seat until steam no longer escapes. The trap is then ready for use and operates as previously described.

This trap is manufactured by the Sarco Fuel Saving and Engineering Company, West Street building, New York City.



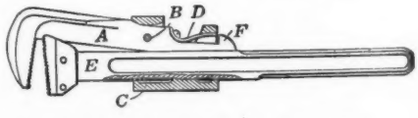
SECTIONAL VIEW OF SARCO STEAM TRAP

Wright Pipe Wrench

The line cut shows a new pipe wrench which is being put on the market by J. F. Wright, Canton, O.

The jaw *A* pivots on the pin *B* in the yoke *C*, which is held against serrations on the under side of the stationary jaw *E* by the spring *D*. While the pin *B* permits the rocking of the jaw *A* in yoke, it does not take any of the strain. This is taken by the lug *F* which is formed on the end of *A* and engages with the yoke.

It is a "one-hand" tool and easily operated.



THE WRIGHT PIPE WRENCH

Morehead Condenser Trap

The new Morehead condenser trap has been especially designed and is employed for use on the return line of vacuum heating systems when a difference in pressure is maintained between ends of the system, the aim being to relieve any steam-heating apparatus of air and water of condension contained therein.

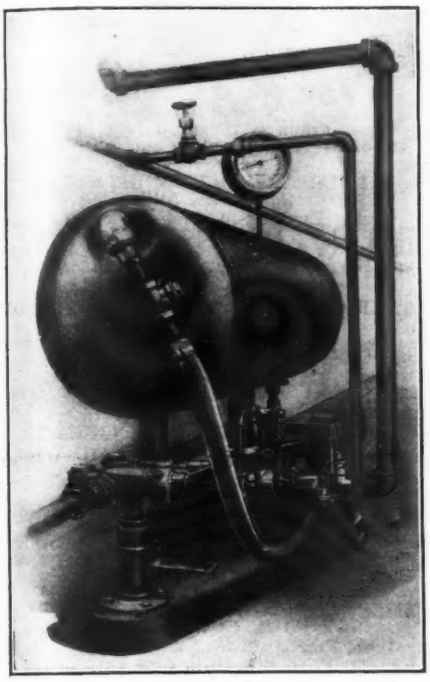


FIG. 1. MOREHEAD CONDENSER TRAP

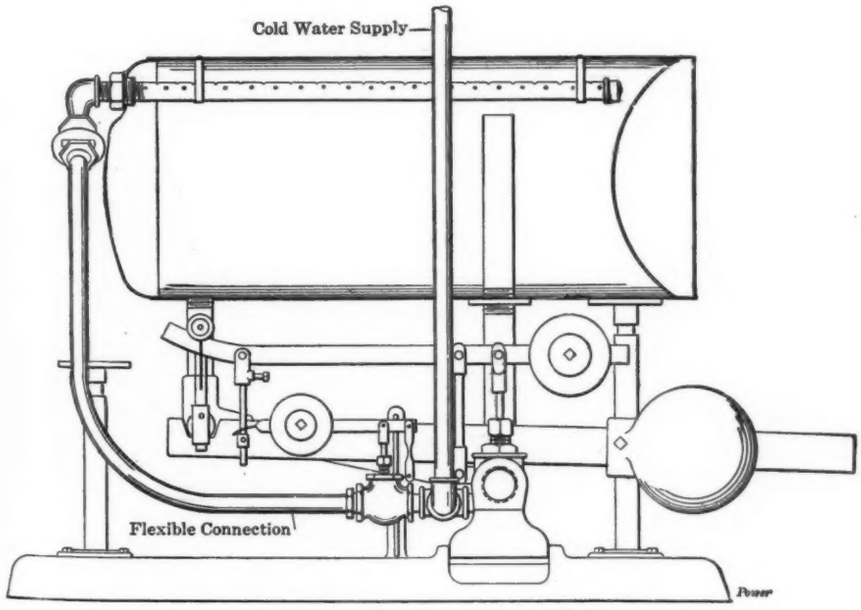


FIG. 2. SECTION THROUGH TRAP

The principle of operation of this condenser trap is extremely simple and similar to that of the return-steam trap of the same make.

A cold-water connection is made through a flexible tube into the tank of the trap, as shown in the illustration. The flexible connection terminates in a perforated pipe inside, and running the entire length of the tank or condensing chamber. An automatically operated valve regulates the intermittent supply of water to the condensing chamber, the cold water striking the steam from the heating system is condensed and thus creates a vacuum. A surprisingly small amount of water is required in the condensing process.

This trap is manufactured by the Morehead Manufacturing Company, Detroit, Mich.

The Leonhardt Safety Water Glass Shield

The accompanying illustration shows the method of application of this shield to a water glass. The shield consists of a metallic sleeve, slotted so that the water level may be seen, and capped at each end with two nurled nuts which form the connection to the water column by means of a rubber gasket and a short section of metallic tubing. The shield prevents glass from flying about the boiler room and injuring the employees in case the gage glass breaks. It also acts as a mechanical protection to the glass and prevents breakage when the glass is being placed in position on the water column.

The sleeve at each end extends into the ends of the glass through the gasket and assists in protecting the glass from the condensed steam which drips from the end of the sleeve instead of running down the inside of the glass tube. The gasket also protects the ends of the glass from being eaten by the action of the water in the glass.

This appliance is made by the Leonhardt Safety Water and Lubricator Glass Shield Company, 167 Dearborn street, Chicago, Ill.

SAFETY WATER GLASS SHIELD

Federal Boiler Inspection Bill

There is a bill before the United States Senate which provides for Federal inspection of the boilers on the locomotives of all common carriers engaged in interstate transportation.

The bill provides for the appointment by the President of an inspector-general and two assistant inspectors-general of locomotive boilers. The area comprised by the various States and Territories is to be divided into 100 inspection districts, each to be covered by an inspector who shall be appointed after competitive examination under the Civil Service rules.

If the bill is passed, it will become effective June 1, 1911.

Demonstration by Lantern Slide

Siemens Brothers Dynamo Works, Ltd., of London, have adopted a practice which might, with advantage, be followed by American manufacturers. They have a collection of one hundred and seventy different lantern slides illustrative of their product and its application which are loaned free of charge for lectures, demonstrations, etc. Many societies have lanterns in their meeting rooms and there is no doubt that if such collections of slides were available they would be very much in demand and used to the mutual advantage of the users and the manufacturers of the apparatus illustrated.

SOCIETY NOTES

The United Steam Engineers of Detroit, which organization is composed of National Association of Stationary Engineers' associations Nos. 1 and 7, International Union of Steam Engineers Nos. 5 and 324, Universal Craftsman's Council of Engineers No. 2, Marine Engineers Beneficial Association No. 4, with about 1400 members in all, have taken up with the University of Michigan the possibility of having a resident instructor at Detroit to devote his time to the organizations, giving lectures and engineering instruction at the association head-

quarters. On October 23 a meeting of the educational committee of the United Steam Engineers was held, at which the entire faculty of the University of Michigan, engineering department, were present, and an important step was taken toward promoting the extension work of the university along the lines suggested above.

PERSONAL

Frank Salomon, formerly chief engineer of Alfred H. Schütte, with headquarters at Cologne, Germany, has accepted an important position with the Otto Gas Engine Works, Philadelphia, Penn.

H. B. Prout, formerly with the Westinghouse Machine Company, in the turbine department, has recently allied himself with the Turbine Equipment Company, 30 Church street, New York City.

NEW PUBLICATIONS

The National Electric Light Association is now issuing to its members the annual *Proceedings*. These two volumes include the report of the thirty-third convention held at St. Louis last May, and mark the twenty-fifth anniversary of the association. The volumes include 2070 pages and 483 illustrations. Among the latter are a fine portrait of President Frueauff and several elaborate colored engravings in the paper on "Street Arcs," by Mr. Ryan. The *Proceedings* embrace some seventy papers and committee reports, all of which are carefully indexed and an abstract is given of each paper or report for ready references. A year ago the edition was 4500 sets. This year, owing to the rapid increase in membership, the edition is not less than 7000 sets, and it is interesting to know that these represent about twenty-eight tons of printed matter.

"The Institute of Operating Engineers" is the title of a 6x9-inch pamphlet setting forth the reasons for the organization of the institute, giving a list of the temporary officers and committees and a draft of the proposed constitution.

In the text there is a great deal of interesting information relating to membership, initiation fees, annual review, central and district organization, executive committee and council, proposed methods of instruction and training of junior members and apprentices, also extracts from the editorials of the leading engineers' periodicals. A copy should be in the hands of every operating engineer in the country. It is furnished on request by M. W. Rice, temporary secretary, 29 West Thirty-ninth street, New York City.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

INTERNAL COMBUSTION ENGINE. William Johnson, Montague, Mich., assignor of one-half to George H. Nelson, Whitehall, Mich. 976,180.

MOTOR. Carl W. Weiss, New York, N. Y. 976,236.

TURBINE. Charles E. Hicks, Mechanicsville, N. Y. 976,260.

ROTARY ENGINE. Lee O. Gilliland, Chicago, Ill., assignor of one-half to Reinhold Reichardt, Chicago, Ill. 976,355.

ROTARY ENGINE. Charles Andrew Bender, Harrisburg, Penn. 976,539.

POWER GENERATOR. Leander J. Cavanaugh and Edward T. Young, Boston, Mass. 976,547.

GAS ENGINE. Edgar F. Prall, New York, N. Y. 976,691.

MOTOR. Charles L. Wilkins, Columbus, Ohio. 976,763.

INTERNAL COMBUSTION ENGINE. Peter M. Allmon, Parker, Ind. 976,770.

ENGINE. Peter W. Kane, Detroit, Mich. 976,808.

BOILERS, FURNACES AND GAS PRODUCERS

LIQUID-FUEL BURNER. William Scrimgeour, Portsmouth, Va., assignor to Mircs Fuel-Oil Equipment Company, Norfolk, Va. 976,221.

LIQUID-FUEL BURNER. Benton Moore, Cherryvale, Kan. 976,592.

LIQUID-FUEL BURNER. William R. McHill, Moran, Kan., assignor of one-third to James H. Cramer, Moran, Kan. 976,820.

POWER-PLANT AUXILIARIES AND APPLIANCES

VALVE. Benjamin F. Silliman, Cleveland, Ohio, assignor to the Cleveland Steel Tool Company, Cleveland, Ohio, a Corporation of Ohio. 976,222.

CONDENSER SYSTEM. Louis R. Alberger, Greenwich, Conn., assignor to Alberger Condenser Company, New York, N. Y., a Corporation of New York. 976,245.

CENTRIFUGAL PUMP. David Wesley Blair, New York, N. Y. 976,340.

PACKING FOR STEAM TURBINES. Gustav Huhn, Berlin, Germany. 976,369.

CENTRIFUGAL PUMP. Rudolf Salzer, Trenton, N. J., assignor to the De Laval Steam Turbine Company. 976,400.

VALVE. William S. Fairhurst, New York, N. Y. 976,449.

VALVE. Johnston Nolan, Philadelphia, Penn. 976,489.

PISTON PACKING. Robert A. Fowden, Philadelphia, Penn. 976,563.

LUBRICATOR. John Andrew Coppers and Eldredge Early Booth, Denison, Texas. 976,637.

VALVE. Guido Schuster, Beszterce, Austria-Hungary. 976,702.

VALVE REGULATOR. Thomas M. Wilkins, East Randolph, N. Y., assignor to Ziermore Regulator Company, Johnsonburg, Penn., a Corporation of Pennsylvania. 976,713.

AUTOMATIC CUTOFF. William C. Coleman, Wichita, Kan. 976,724.

ENGINE VALVE. Rudolf Wintzer, Milwaukee, Wis., assignor to Power and Mining Machinery Company, New York, N. Y., a Corporation of New Jersey. 976,766.

SPARKING PLUG FOR INTERNAL COMBUSTION ENGINE. Robert Frederick Hall, Moseley, Birmingham, England. 976,797.

VALVE. James H. Hulings, Parsons, W. Va., assignor of one-half to Wade H. Miller, Parsons, W. Va. 976,804.

CONTAINER FOR LUBRICANTS. Adolf Nemecek, Josefstadt, Austria-Hungary. 976,825.

ELECTRICAL INVENTIONS AND APPLICATIONS

DYNAMO ELECTRIC MACHINE. Robert B. Williamson, Milwaukee, Wis., assignor to Allis-Chalmers Company, a Corporation of New Jersey. 976,420.

ELECTRIC SWITCH. John Hastings Wooll, San Francisco, Cal. 976,422.

SWITCH. Herbery W. Cheney, Norwood, Ohio, assignor to Allis-Chalmers Company, a Corporation of New Jersey, and the Bullock Electric Manufacturing Company, a Corporation of Ohio. 976,549.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

INTERNAL COMBUSTION ENGINEERS' ASSOCIATION.

Pres., Arthur J. Frith; sec., Charles Kratsch, 416 W. Indiana St., Chicago. Meetings the second Friday in each month at Fraternity Halls, Chicago.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; acting sec., Charles P. Crowe, Ohio State University, Columbus, Ohio. Next meeting, Youngstown, Ohio, May 18 and 19, 1911.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.